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Final Report

CWR 700-9-II

TESTING OF TRANSPIRATION AIR COOLED TURBINE BLADES

VOLUME II

Report on Item Ic
Contract NOas 58-404-c

Curtiss-Wright

Published May 31, 1960

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CWR REPORT NO. 700-9-II

Department of the Navy
Bureau of Weapons

Contract NOas 58-404c

Security Control No. RE 60-7-14

Copy No. ___/___

CWR Report No. 700-9-II

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SUMMARY

An evaluation test of a typical transpiration cooled turbine blade was performed on the combustion chamber sector rig, prior to engine testing, in order to evaluate the coolant requirements of high temperature operation. A total of one hour and forty-five minutes at gas temperatures of 2100°F or higher was accumulated with two porous test blade assemblies during this combustor sector rig testing. Of this time, the blades were tested for fifty-one minutes at gas temperatures of 2800°F. The results obtained demonstrate the ability of the blades to operate in an environment of turbine inlet temperatures of 2500°F and higher.

Cascade tests were conducted to determine the contamination effects of JP-4 exhaust products on porous blade airfoils. These tests revealed, after fifty hours of test, that no significant contamination occurred for a porous material with a permeability of $K/t = 5 \times 10^{-4}$ ft. in a temperature environment of 1600°F.

Full scale engine tests of transpiration cooled turbine rotor and stator blades were successful up to average turbine inlet temperatures just under 2200°F. Inability to complete testing up to 2500°F was the result of premature and unexpected failure of an engine test rig component which destroyed the utility of the turbine rotor stage. In spite of the test rig failure the excellent condition of the transpiration air cooled stator blades, which had been subjected to temperature peaks estimated to be between 2800°F to 3000°F, and inspection of the undamaged portions of rotor blades, indicate that operation at average turbine inlet temperatures of 2500°F is possible.

Cooling air flow to the rotor blades was 4% as prescribed in the design (2500°F) requirements which test results showed was more than sufficient as overcooling was evident on some areas of the blades. In the stator blades less than 4% cooling air flow was used in the tests conducted, with temperature peaks experienced up to 3000°F. Based on these results it is indicated that less then 4% cooling air may be required for operation at an average of 2500°F.

With the prescribed cooling air flows the designed maximum skin and strut temperatures of 1600°F and 1200°F, respectively, were not exceeded as indicated by material conditions observed after full scale engine testing.

CONCLUSIONS

1. Contamination of the porous skin airfoils by JP-4 exhaust products is not significant enough to appreciably affect the coolant flow.

2. The engine test results, wherein stator blades operated satisfactorily in temperature environments in the range of 2800°F to 3000°F (peak temperatures), indicate that the feasibility of transpiration air cooled stator blades is established.

3. The demonstration of the feasibility of employing a transpiration air cooled rotor stage up to 2500°F average turbine inlet temperature was not accomplished, because of test rig failure. However, inspection of the rotor blades after operation at 2200°F clearly indicates the ability of this type of cooled blade to operate in temperature environments of 2500°F, and above.

4. Hot gas flow tests on a transpiration air cooled blade configuration indicates that the heat-transfer theory used in the design of the blades is adequate.

RECOMMENDATIONS

1. Redesign turbine shroud of test vehicle.

2. Revise location of compressor bleed air for stator coolant air flow.

3. Continue the engine test program to demonstrate operation at 2500°F average turbine inlet temperature.

DISCUSSION

A. Transpiration Cooled Blade Rig Test

An evaluation test of a typical transpiration cooled turbine blade was performed on the combustion chamber section rig in order to evaluate the coolant demand at high temperatures.

A transpiration cooled turbine blade was installed on the rig with the upstream coolant pressure set at 35 psi. The downstream pressure (static pressure of gas around blade) was, in this test, the ambient pressure. Readings of coolant upstream pressure level and volume flow were taken for twenty-five separate points of Δ P across the skin varying from 35 psi to zero. The coolant was ambient air at a temperature of 40°F. Table VI-I develops the calculation of the actual weight flow at each point.

Figure VI-I shows the variation of the average effusion rate with across the blade wall. This average was obtained by dividing the total actual weight flow to the blade (lb/sec) by the total porous surface area. The calculated wall temperature for the new blade skin varies between 1350°F and 1500°F for effusion rates between .1 and .3 lb/sec-ft^2. This curve is convenient for rapid determination of the upstream coolant pressure levels required to obtain a given effusion rate.

Wall temperatures were approximated by comparing the color of the blade during test with color charts for alloy steel. The values of temperature so determined compare favorably with theoretical calculation runs made during the heat transfer design. For example, the metal color at point 17, Table VI-I, corresponded to a temperature of 1050°F on the color chart. At this point the average effusion rate was .220 lb/sec-ft^2, Table VI-II. This rate through a porous surface at $k/t = 1 \times 10^{-10}$ feet and .031 inches thickness would yield temperatures between 1450°F and 1550°F. However, the air in the rig test was at least 350°F cooler than it would be in the engine, which reaffirms the validity of the 1050°F observed.

Coolant bleed above one and one half (1.5) percent in the plot in Figure VI-2 is dotted because the color of the metal surface during test was not noticeably different from what it was at room temperature. At these points comparison of the measured effusion rates to theoretical rates was the only means of determining temperature. The test points at which no color change was apparent were Nos. 1 through 13. The pressure difference across the wall was maximum at Point 1, and was reduced in increments of two (2) to two and one half (2.5) pound per square inch. It may well be that the metal temperature drops more sharply than is indicated by the dotted part of the curve in Figure VI-2. An expeditious way of determining the metal temperature in this area would be by installing a thermocouple in the strut. Despite the fact that there may be a 200 - 300°F difference between the strut and skin temperatures, the measured strut temperatures would give positive indication of any major changes in the skin temperature.

Figure VI-3 is a graphical determination of the exponent "n" in the relationship, $\rho V = C_k(P_i^2 - P_e^2)^n$, where ρV is the effusion rate (G) mentioned previously. The value of .589 obtained is reasonable as compared to the .625 value suggested by the manufacturers of the porous material "Poroloy." Figure VI-4 is a drawing of the test blade assembly and Figures VI-5 and VI-6 are photographs of two blade assemblies after test.

B. Porous Blade Contamination Test

Tests were conducted on transpiration cooled turbine blades to determine the contamination of JP-4 exhaust products on the porous blade airfoil. These tests were performed on a small-scale burner rig simulating engine conditions as closely as possible with the exception of rotation.

A total of sixty-one hours of testing was conducted with fifty hours of endurance testing at $1600^{\circ}F$ on one blade with no cooling air being passed through the porous airfoil. Flow checks were made on the blade at the beginning of the test and at intervals of ten hours of running or less. Initially flow checks were made at smaller intervals of time, but changes in flow rate due to any contamination, if any, that the equipment utilized to measure flow could not detect any changes. Even the flow checks measured at the larger intervals of test time, normally ten hours, indicated only slight decreases which were quite close to the limits of accuracy of the recording instruments.

Results of the data obtained are plotted on Figure VI-7. As can be seen on Figure VI-7(a), the loss in pressure drop through the porous skin was approximately 2.8 inches of water, after fifty hours of test, which is considered negligible. Design of the metering orifices for a transpiration cooled turbine blade cannot be held to this accuracy. The loss in air flow resulting from what contamination did occur in the fifty hours is plotted on Figure VI-7(b). The loss in flow indicated is such a minute quantity that it can be neglected as variations in the permeability of porous material from production runs would result in greater flow rates changes from piece to piece.

The results obtained in testing porous airfoils in the exhaust products of JP-4 fuel, at $1600^{\circ}F$ for "Poroloy" material with a permeability of $K/t = 5 \times 10^{-10}$ feet, it appears that contamination of the transpiration air-cooled porous airfoil is negligible.

C. Full Scale Engine Test

Testing of the transpiration air cooled turbine stator and rotor blades was directed toward demonstrating the feasibility of transpiration air cooled turbine blades at an elevated turbine inlet temperature averaging 2500°F.

1. Compressor Calibration

Prior to testing the transpiration cooled turbine a calibration of the modified compressor was necessary to ascertain that it would match the redesigned turbine. The standard J-65 two stage solid blade turbine was used to conduct this test. Although, this combination of the standard turbine combined with the modified compressor were not matched to each other, they were close enough to satisfactorily check out the compressor. Initial testing indicated that the compressor weight flow was too high. Adjustment of the variable inlet guide vanes, incorporated into the compressor, corrected the weight flow and gave the compression ratio designed to match the new 2500°F turbine.

The only difficulty encountered in the compressor checkout tests was that the normal shaft critical speed that exists at 6000 rpm was moved up closer to the design speed. In order to provide a greater range of speed operation near the new design point, weight was added to the compressor to lower the critical speed back to approximately 6000 rpm.

Instrumentation of the first two compressor stages was made to obtain vibratory stress levels on the blades during test. Stress levels recorded during these tests were well below the allowable levels, and there were no indications of stall or surge in the range of operation anticipated for 2500°F turbine inlet temperature operation.

2. Transpiration Cooled Turbine Blade Testing

Testing of the transpiration air cooled turbine rotor assembly, shown in Figure VI-8,(and the transpiration cooled turbine stator blades) consisted of the program listed in Table VI-III.

The initial tests were conducted to check cooling air flows, temperature distribution in the combustion chamber, and general test rig operation at relatively low turbine inlet temperatures Difficulties were encountered with blades fabricated with the Rigimesh porous airfoil material. Upon attaining a little above 55% design speed one of the Rigimesh airfoils separated from the blade supporting strut. Inspection of the blade revealed that poor brazing existed on the porous material due to oxide films in the porous material fibers. Replacement of the blade was made and further testing was carried on. In the next series of tests conducted loss of another Rigimesh porous airfoil occurred just above 80% speed (6050 rpm). Figures VI-9 and VI-10 shows the blade with the missing airfoil, and the minor resulting secondary damage to the remaining rotor blades. No damage was incurred to the transpiration cooled stator blades or any other engine component. Inspection of this blade also revealed poor bonding of the braze to the Rigimesh porous material, due to oxide films. It was decided at this time to replace all the blades made from Rigimesh material, approximately 50% of the wheel, with blades made from Poroloy material.

During this initial testing it was also noted that the turbine rotor shroud was distorted. The cause of this was believed to be that the fabricating vendor had not properly stress relieved the part after welding. As a result of this distortion higher tip clearances existed.

In the next series of tests with higher turbine inlet temperatures, ranging from 1850°F to 2000°F, the loss of another airfoil occurred. This blade was replaced and testing was continued to design speed, 7460 rpm, with a turbine inlet temperature average of 2100°F or above. This point was set the next day and shortly after stabilizing, an increase in vibration was indicated along with a surge in exhaust gas temperatures. Inspection of the engine after shutdown showed that the stator blades were burned at six points, and had seriously damaged all the rotor blades. Figures VI-11 and VI-12 show the transpiration cooled turbine stator assembly. It can be seen on these photographs that there are six definite burned areas resulting from combustion chamber "hot spots". It also can be noticed on Figure VI-11 the four double cooling air lines supplying the exhaust duct cooling air. These four bleed air lines tie in directly to the location of four of the "hot spots". The other two "hot spots" are also directly in line with the two bleed points for stator blade cooling air. It is quite apparent that these cooling air bleed stations created the six torching areas in the combustion chamber, and starved the secondary air liner cooling supply in these areas.

Further investigation also revealed that the stator air cooling supply was not up to design requirements, as can be seen on Figure VI-24a. Unfortunately late delivery of the stator air pressure boost pump did not permit a verification test of the vendor calibration results, and the pump was used directly.

As a result of the torching condition in the combustion chamber and the burning through of the stator blades the rotor assembly was damaged beyond repair. Figures VI-13 and VI-14 show the damaged incurred on the turbine rotor assembly.

A close-up view of the burned stators is shown in Figure VI-15. Figure VI-16 shows the typical secondary damage done to the remaining stator and rotor blades. It can be noted that the condition of the blades in the areas undamaged by secondary effects is quite good.

Closer observation of the blades revealed that excessive cooling was occurring on the suction side of both the rotor and stator blades. In the theoretical heat transfer analysis discussed in Section I, Volume I of the report the pressure distribution around the blade periphery was determined by the use of an analog. However, due to the fact that this was an entirely new turbine design only one section, the mean section of the blades, was analyzed in this manner for chordwise pressure distribution. As the spanwise pressure profile was assumed to obtain the chordwise pressure distribution without any background on this particular turbine it was felt that the accuracy of any pressure profile determined by the analog method would be highly questionable at any other section than the mean.

In order to improve the cooling air distribution on the blades it was felt that masking of the blade was necessary in the areas that appeared to be receiving more than the required amounts of cooling air. The method evolved to mask these cooler areas was by utilizing a metal spray technique generally known in industry as "metalizing". By this metalizing technique it was possible to mask the desired areas and distribute cooling air to the hotter portions of the blade. Laboratory checks of these metalized areas showed that the areas were still somewhat porous. Again time did not permit a detailed study, but an investigation in the future would be of value to possibly use this method to obtain variable permeability over the blade airfoil surface area. Figure VI-17 shows a stator blade that has been metalized on the suction side at the root of the airfoil.

In order to conduct further testing it was necessary to use the 'Rigimesh' rotor blades. As discussed previously all of these blades had been removed from the rotor assembly as the bond to the porous airfoil was questionable due to the dirtiness of the porous material. Plug welding of the porous airfoil skin to the blade strut was attempted as a means of salvaging these blades, as shown in Figure VI-18. It can be seen in this figure that the plug welding created dimples in the airfoil surface that would disturb the turbine aerodynamics. However, it was felt at this time that this was the only way further testing could be conducted. Although engine performance would suffer, the objective of demonstrating the feasibility of transpiration cooled turbine blades at elevated turbine inlet temperatures could still be attained. Metalizing at the root section on the suction side of the rotor blade was also done to improve cooling air distribution, and is also shown in Figure VI-18.

Another modification made was to utilize external cooling air for the stator blades and exhaust duct to eliminate the uneven mass flow through the combustion chamber, basically the six extremely "hot spots" which caused the failure of the stator blades. This could have been corrected by modifying the method of bleeding this cooling air; but again, time did not allow the incorporation of the necessary modifications for better distribution of cooling air bleed. The source of external cooling air bleed was portable 600 FM constant displacement compressors.

The engine test program was terminated because as the rated speed point was being set at approximately 2200°F turbine inlet temperature, engine surge was noted, indicating an engine component had failed. Inspection of the engine, after shut-down, showed the turbine rotor shroud had broken loose from the engine casing, and it was resting (wrapped around) the turbine rotor stage. The turbine blade tip areas showed signs of severe tip rub. The airfoil skins of two turbine rotor blades were completely torn loose, and damage was noted on the remaining airfoils. An analysis of the damaged rotor, Figure VI-19, shows that the blades were running relatively cool, and discounting the secondary damage, were in excellent condition considering the severe conditions encountered. Figure VI-20 shows the turbine shroud, after it was removed (unwrapped) from the rotor stage. Analysis of the part indicated failure was caused by, primarily, faulty welding of the shroud. While this was the immediate cause of the failure, the extreme distortion of the shroud noted in previous running leads to the belief that the true cause of failure was the improper heat treatment of the shroud. As the temperature level of the engine was increased, the distortion increased, until finally the attaching lugs failed, loosening the shroud from the engine casing.

Inspection also revealed that the condition of the transpiration cooled stator blades after these tests was excellent, and they only suffered minor secondary damage as can be seen in Figure VI-21. The effectiveness of the masking of the blades at the root section on the suction side was definitely proven by the condition of the blades by the apparent change in cooling air flow to the more critical areas of the blade airfoil surfaces.

Analysis of data obtained during the testing of the transpiration cooled turbine blades show that the average turbine inlet temperatures attained were over 2100°F as shown on Figure VI-22, along with the exhaust gas temperatures recorded. Tests were also conducted without a variable area nozzle and the turbine inlet and exhaust gas temperatures obtained during these runs are plotted on Figure VI-23. The basic purpose of the test with a wide open exhaust duct was to check out basic engine test rig operation.

The total test time accumulated was 11 hours and 47 minutes of which one hour was above 1800°F and thirty-five minutes were between 2000°F and 2200°F.

The cooling air flows to the rotor and stator blades are plotted on Figure VI-24. As can be seen on Figure VI-24a the stator cooling air supplied through the boost compression is below the required design coolant flow of 4%. Also in this figure is the quantity of external coolant air flow used in the latter tests after discontinuation of engine bleed air. Figure VI-24b shows the rotor blade cooling air flow. As can be seen the quantity of air flowing to the rotor blades was very close to the prescribed design cooling air flow to the rotor blades. Although the external cooling air flow to the stator blade was not to the percentage design requirements the difference is made up by the fact that the external cooling air is at a much lower temperature as shown on Figure VI-25. Figure VI-25 also shows the temperature of all the various air temperatures in the engine test rig. It can be seen that the temperature rise of the stator cooling air is greater than the rotor cooling air due to work performed by the boost compressor in increasing pressure.

The compressor weight flow obtained during the transpiration cooled turbine blade testing is shown in Figure VI-26. As discussed previously the compressor was tuned to match the transpiration cooled turbine design, and apparently came quite close from the limited data obtained. Unfortunately, loss of instrumentation during testing limited the amount of engine performance data sufficiently so that a thorough engine performance analysis could not be made. Also the fact that the excessive tip clearances, created by the turbine shroud distortion, that occurred in the early phases of testing would give results unfavorable to overall engine test rig performance. Figure VI-27 does show some of the basic engine data obtained during these tests.

Inspection of the hot section components of the test vehicle at the conclusion of the test program revealed that except for the turbine shroud, all parts were in excellent condition. In particular, the combustion chamber showed only slight metal discoloration in spite of the 3000°F peak temperatures that must have existed during the first 2200°F test point.

The most significant metal temperature measurements were obtained by painting the pertinent areas with Thermocolor temperature indicating paints. Temperature measurements made by this technique showed the combustion chamber inner and outer liners did not exceed 1200°F; the stator supports, and stator blade airfoil root and tip areas did not exceed 824°F; and the turbine rotor blade airfoils root sections did not exceed 824°F. These measurements were noted at the 2200°F average turbine inlet temperature test points.

Overall operation of the engine test vehicle was exceptionally smooth. In particular, the transpiration air cooled turbine configuration started and accelerated very smoothly. Maximum exhaust gas temperatures, on starting, did not exceed 950°F with the open exhaust duct configuration; and did not exceed 1350°F with the variable area nozzle configuration. Response of the engine to changes in settings (RPM and turbine inlet temperature) was good and no instabilities were noted up to the 2200°F test point.

Point Number	Coolant Pressure P_1 (psia)	Actual Coolant Density ρ_{ACT} ($\frac{lbs}{ft^3}$)	$\frac{\rho_{ACT}}{\rho_{STD}}$	$\left[\frac{\rho_{ACT}}{\rho_{STD}}\right]^{\frac{1}{2}}$	Volume Flow W_{STD} (FT^3/MIN)	Weight Flow ($\frac{lbs}{sec}$)	
						$Q_{STD} = W_{STD} \cdot \rho_{STD}$	$Q_{ACT} = Q_{STD} \sqrt{\ }$
1	48.5	.261	3.488	1.867	17.0	.0212	.0396
2	46	.248	3.314	1.820	16.5	.0206	.0375
3	43.5	.234	3.127	1.768	16.0	.0199	.0352
4	40.5	.218	2.913	1.707	15.5	.0193	.0329
5	38.5	.208	2.779	1.667	15.0	.0187	.0312
6	35.5	.191	2.553	1.598	14.5	.0181	.0289
7	33.5	.181	2.419	1.555	14.0	.0175	.0272
8	31.5	.170	2.272	1.507	13.5	.0168	.0253
9	30.5	.164	2.192	1.481	13.0	.0162	.0240
10	28.5	.154	2.058	1.435	12.5	.0156	.0224
11	27.5	.148	1.978	1.406	11.0	.0137	.0193
12	25.5	.137	1.831	1.353	10.5	.0131	.0177
13	24.5	.132	1.764	1.328	10.0	.0125	.0166
14	23.5	.127	1.697	1.303	9.5	.0118	.0154
15	22.5	.121	1.617	1.272	8.5	.0106	.0135
16	21	.113	1.510	1.229	8.0	.0100	.0123
17	20	.109	1.456	1.207	7.5	.0094	.0113
18	19.5	.105	1.403	1.184	7.0	.0087	.0103
19	18.5	.100	1.336	1.156	6.5	.0081	.0094
20	17.5	.094	1.256	1.121	6.0	.0075	.0084
21	17	.092	1.229	1.109	4.5	.0056	.0062
22	16	.086	1.149	1.072	4.0	.0050	.0054
23	15.5	.084	1.123	1.060	3.5	.0044	.0047
24	14.5	.078	1.042	1.021	2.5	.0031	.0032
25	13.5	.073	.097	.31	0	- -	- -

<u>Table VI-I</u> Tabulation of Coolant Flow Parameters - Open Tipped Test Blade. $\rho_{STD} = .07482 \frac{LBS}{FT^3}$; $T_{ACT} = 500°R$

Point Number	Coolant Pressure $\left(\frac{LBS}{FT^2}\right)_{ABS}$ P_i	P_i^2 $\left(\frac{LBS^2}{FT^4}\right)$	$P_i^2 - P_e^2$ $\left(\frac{LBS^2}{FT^4}\right)$	$\frac{P_i^2 - P_e^2}{\tau}$ $\left(\frac{LBS^2}{FT^5}\right)$	Effusion Rate $G\left(\frac{LBS}{SEC,FT^2}\right)$	% Bleed (BASED ON J-65 PARAMETERS) $Q = 19$ [illegible]
1	6984	48,776,256	44,997,120	25,712	.770	3.51
2	6624	43,877,376	40,098,240	22,913	.73	3.33
3	6264	39,237,696	35,458,560	20,262	.685	3.12
4	5832	34,012,224	30,233,088	17,276	.641	2.92
5	5544	30,735,936	26,956,800	15,404	.608	2.77
6	5112	26,132,544	22,353,408	12,773	.563	2.56
7	4824	23,270,976	19,491,840	11,138	.53	2.41
8	4536	20,575,296	16,796,160	9,598	.492	2.24
9	4392	19,289,664	15,510,528	8,863	.466	2.13
10	4104	16,842,816	13,063,680	7,465	.436	1.99
11	3960	15,681,600	11,902,464	6,801	.376	1.71
12	3672	13,483,584	9,704,448	5,545	.344	1.57
13	3528	12,446,784	8,667,648	4,953	.323	1.47
14	3384	11,451,456	7,672,320	4,384	.299	1.37
15	3240	10,497,600	6,718,464	3,839	.262	1.20
16	3024	9,144,576	5,365,440	3,066	.239	1.09
17	2880	8,294,400	4,515,264	2,580	.220	1.00
18	2808	7,884,864	4,105,728	2,346	.200	.91
19	2664	7,096,896	3,317,760	1,896	.183	.83
20	2520	6,350,400	2,571,264	1,469	.164	.745
21	2448	5,992,704	2,213,568	1,265	.121	.55
22	2304	5,308,416	1,529,280	874	.105	.48
23	2232	4,981,824	1,202,688	687	.092	.42
24	2088	4,359,744	580,608	332	.062	.28

Table VI-II Tabulation of Calculated Data for Plotting Figure VI-3

P_e = Barometer = 13.5 psi = 1944 psf

τ = Skin Thickness = .021 inches = .00175 feet

Table VI-III

Cooled Turbine Blade Engine Test

	Type Test	Conditions	Remarks
1.	Compressor Calibration	a. 3000 RPM b. 4000 RPM c. 5000 RPM d. 6000 RPM e. 7000 RPM f. 7460 RPM (Design RPM)	Compressor weight flow too high. Reset variable inlet guide vanes to obtain reduced flow.
2.	Compressor Calibration	a. Same as (1)	Compressor weight flow still too high.
3.	Compressor Calibration	a. Same as (1)	Compressor weight flow and pressure ratio tuned to predicted match point for cooled turbine.
4.	Transpiration Cooled Turbine	a. Starting b. 2500 RPM c. 3420 RPM d. 4000 RPM	Engine started cool. Inspected blades on stand. Good Condition.
5.	Transpiration Cooled Turbine	a. Start b. 4000 RPM c. Acceleration	Normal cool start. Noticed spark in exhaust. Found Rigimesh turbine blade airfoil missing. All other blade all right. Replaced with Poroloy blade.
6.	Transpiration Cooled Turbine	a. Start b. 4000 RPM c. 6075 RPM d. 6050 RPM	Normal cool start. Inspected blades on stand. Good Condition. Check point. Vibration increased. Shut down. Rigimesh airfoil starting to come off. Removed all Rigimesh blades and installed all Poroloy blades. Found shroud distorting.

	Type Test	Conditions	Remarks
7.	Transpiration Cooled Turbine	a. Start	Normal cool start.
		b. 4000 RPM	
		c. 6000 RPM	Inspect blades. Good condition.
		d. 6550 RPM	Inspect blade. Good condition.
		e. 6820 RPM	Inspect blades. Good condition. 1850°F turbine inlet temperature. Install cooling water spray on exhaust.
8.	Transpiration Cooled Turbine	a. Start	Normal cool start.
		b. 4000 RPM	
		c. 7050 RPM	1975°F turbine inlet temperature. Vibration up after few minutes. Inspected blades. Found airfoil missing and three others damaged.
9.	Transpiration Cooled Turbine	a. Start	Normal cool start.
		b. 4000 RPM	
		c. 7460 RPM	Design speed. Turbine inlet temperature above 2100°F. Developed oil leak. Inspected blades. Good condition.
10.	Transpiration Cooled Turbine	a. Start	Normal.
		b. 7460 RPM	Vibration increased shortly after setting speed. Inspected blades and found stators burned through at six (6) hot spots damaging rotors. Replaced all stator and rotor blades.
11.	Transpiration Cooled Turbine	a. Start	Normal.
		b. 4000 RPM	
		c. 6500 RPM	Inspected blades. Good condition.
		d. 7700 RPM	Overspeed condition. Inspected blade. Good condition.
12.	Transpiration Cooled Turbine	a. Start	Normal.
		b. 4000 RPM	
		c. 6000 RPM	Inspected blades. Good condition.
		d. 7000 RPM	
		e. Acceleration	E.G.T. surged. Shut down. Inspected blades. Found turbine shroud failure at weld. Damaging rotor blades beyond repair. End of test.

Total testing time: 11 hours and 47 minutes

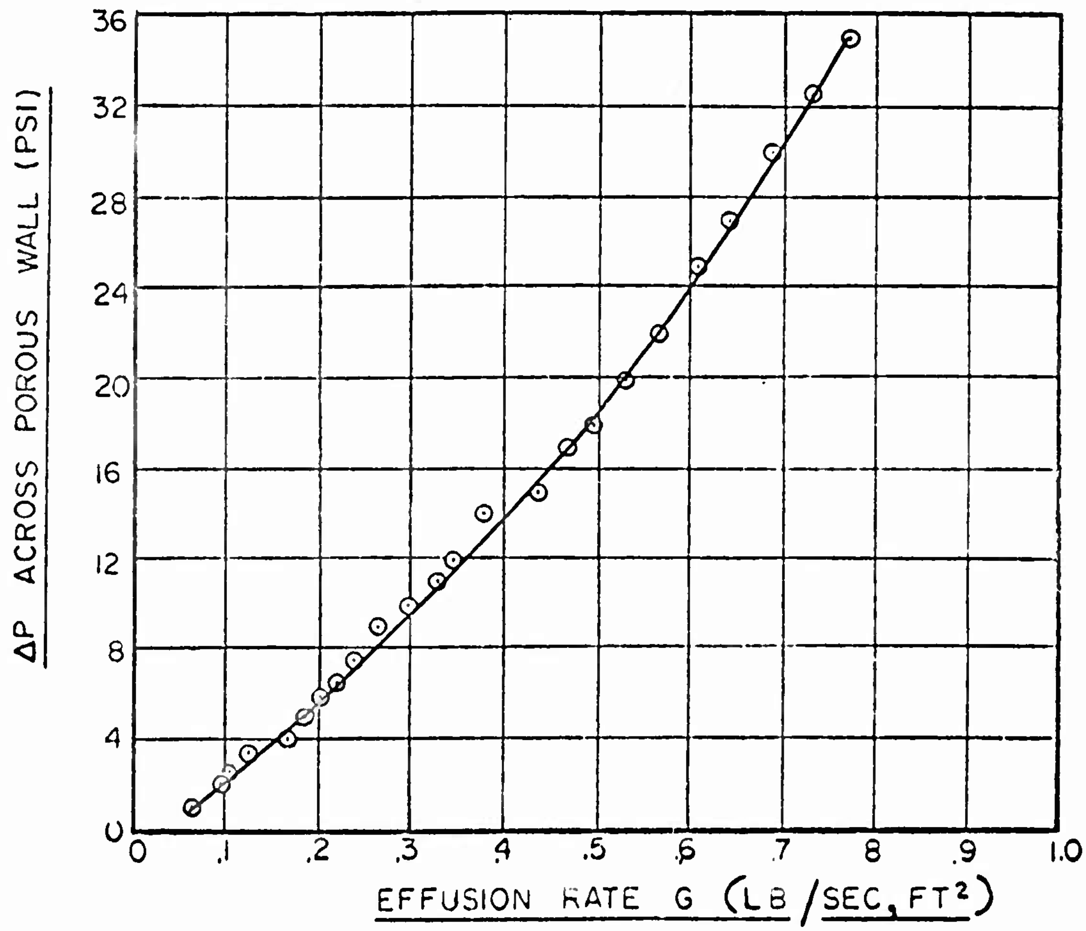


FIGURE III-1 AVERAGE COOLANT EFFUSION RATE VS PRESSURE DIFFERENCE ACROSS TEST BLADE SKIN

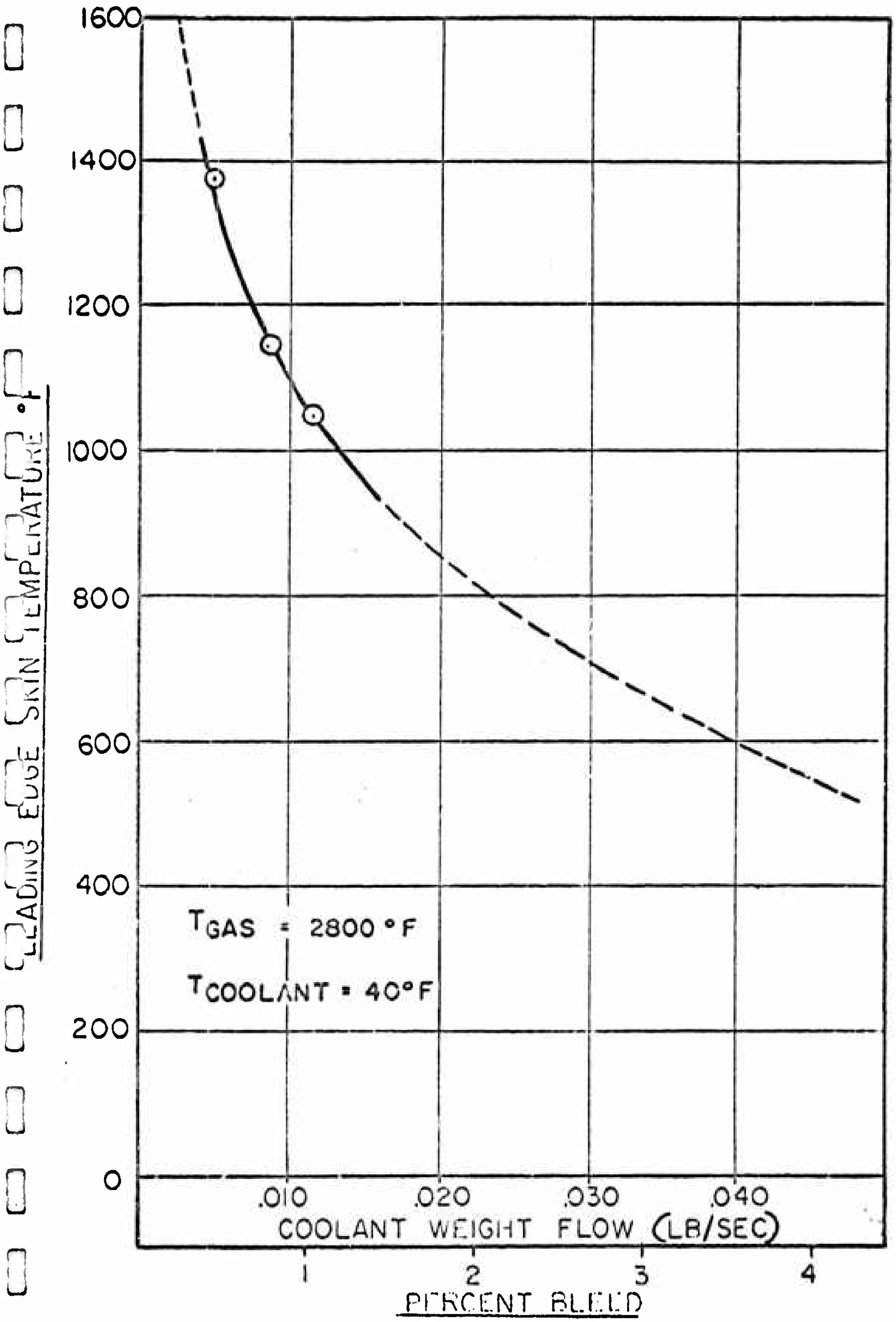


FIGURE VI-2 VARIATION OF LEADING EDGE TEMPERATURE AT MIDSPAN WITH COOLANT WEIGHT FLOW

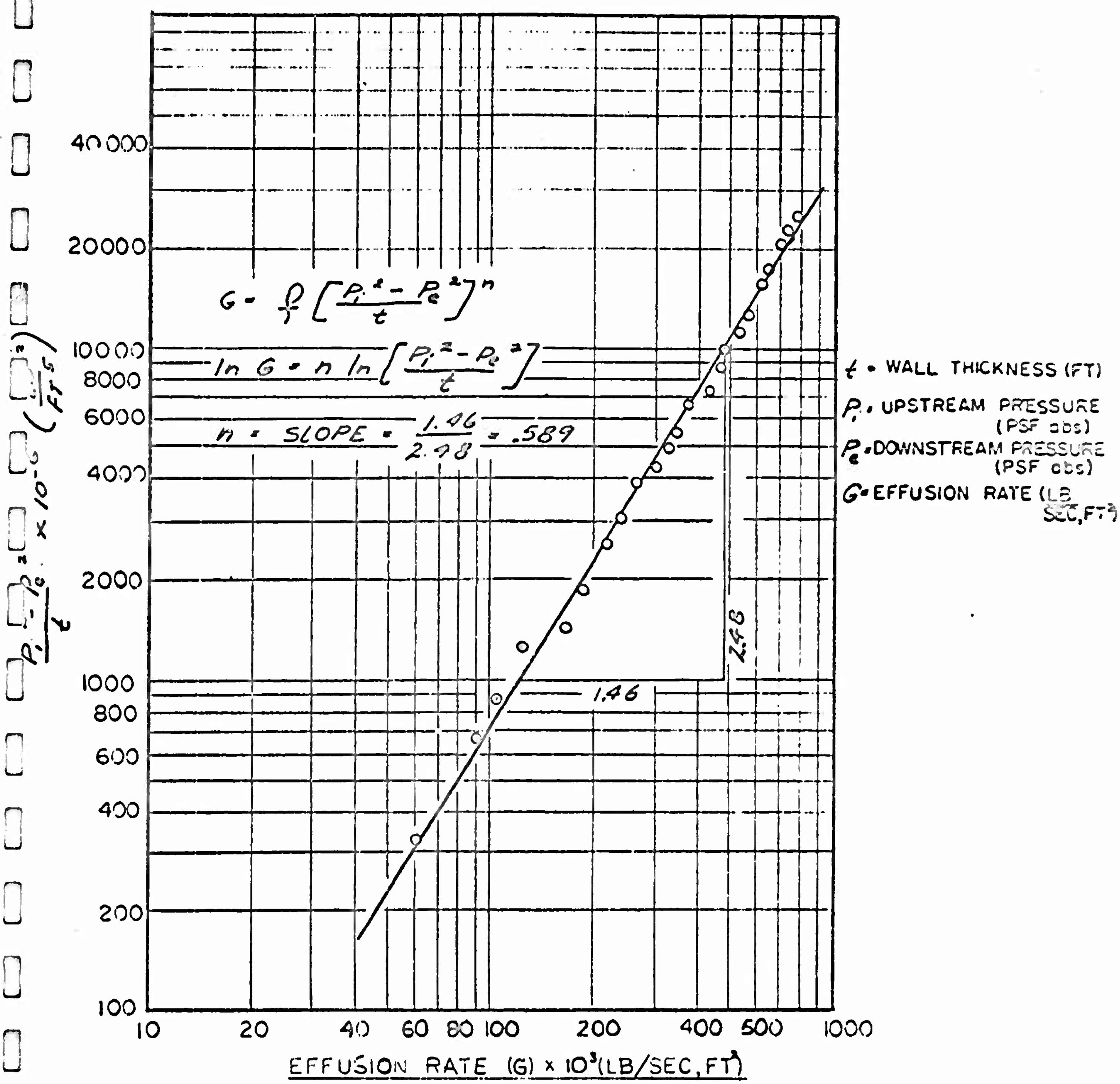


FIGURE VI-3 GRAPHIC DETERMINATION OF EXPONENT "n" IN BASIC EQUATION DESCRIBING FLOW OF AIR THROUGH POROUS MEDIA

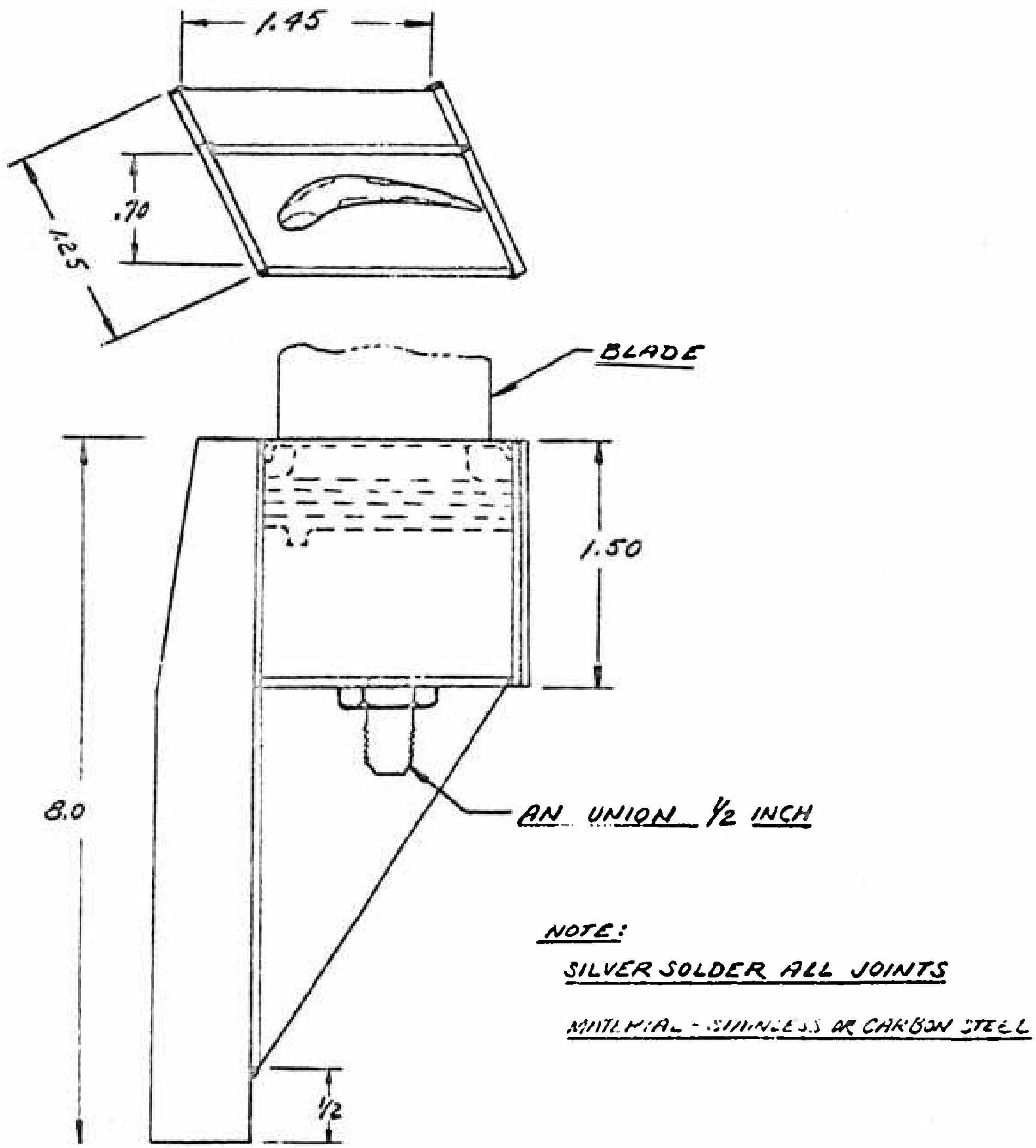


FIG. VI-4 TRANSPIRATION AIR COOLED BLADE TEST ASSEMBLY FOR EVALUATION IN COMBUSTION CHAMBER SECTOR RIG

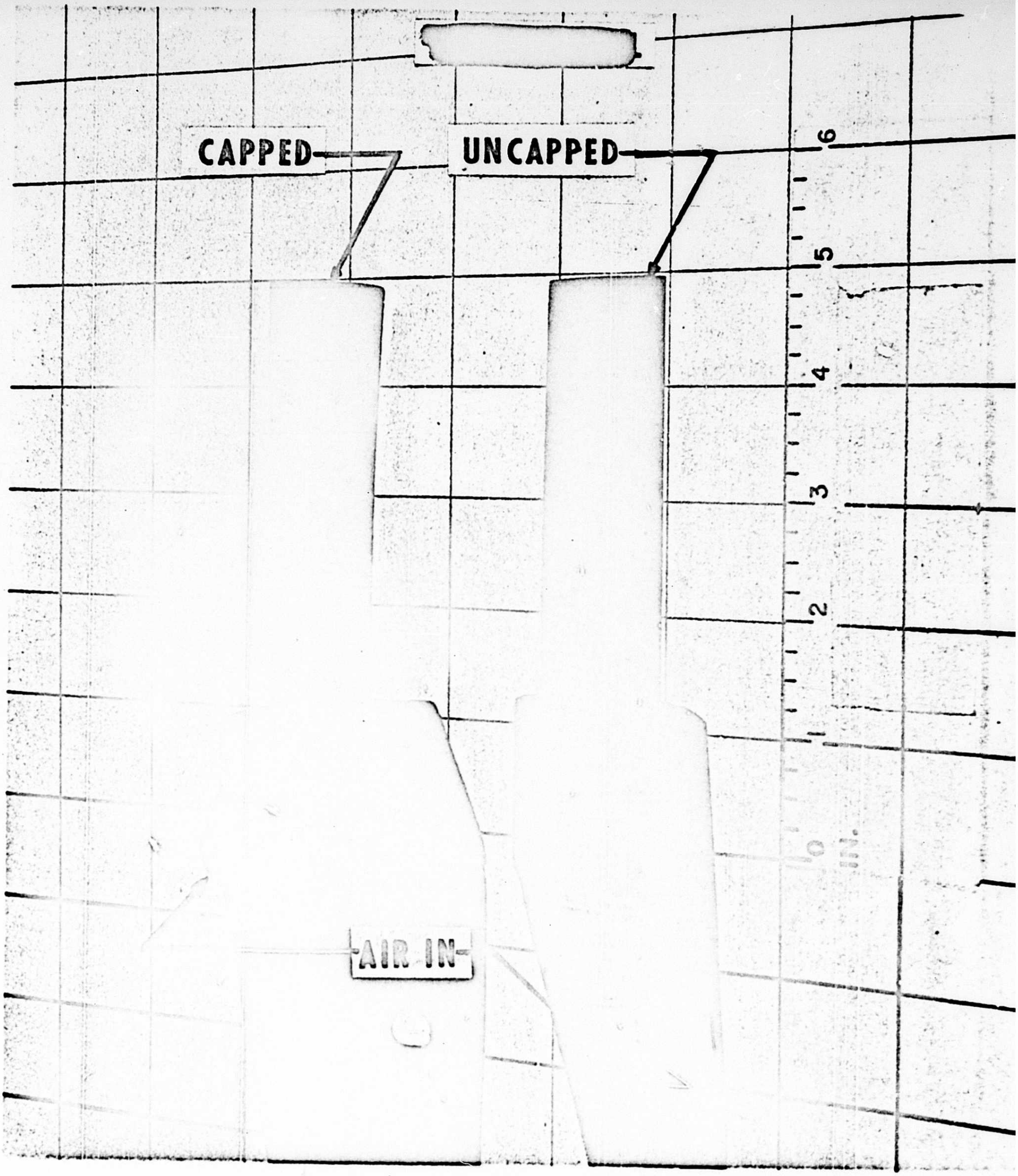


FIG.5—BLADE ASSEMBLIES AFTER TEST

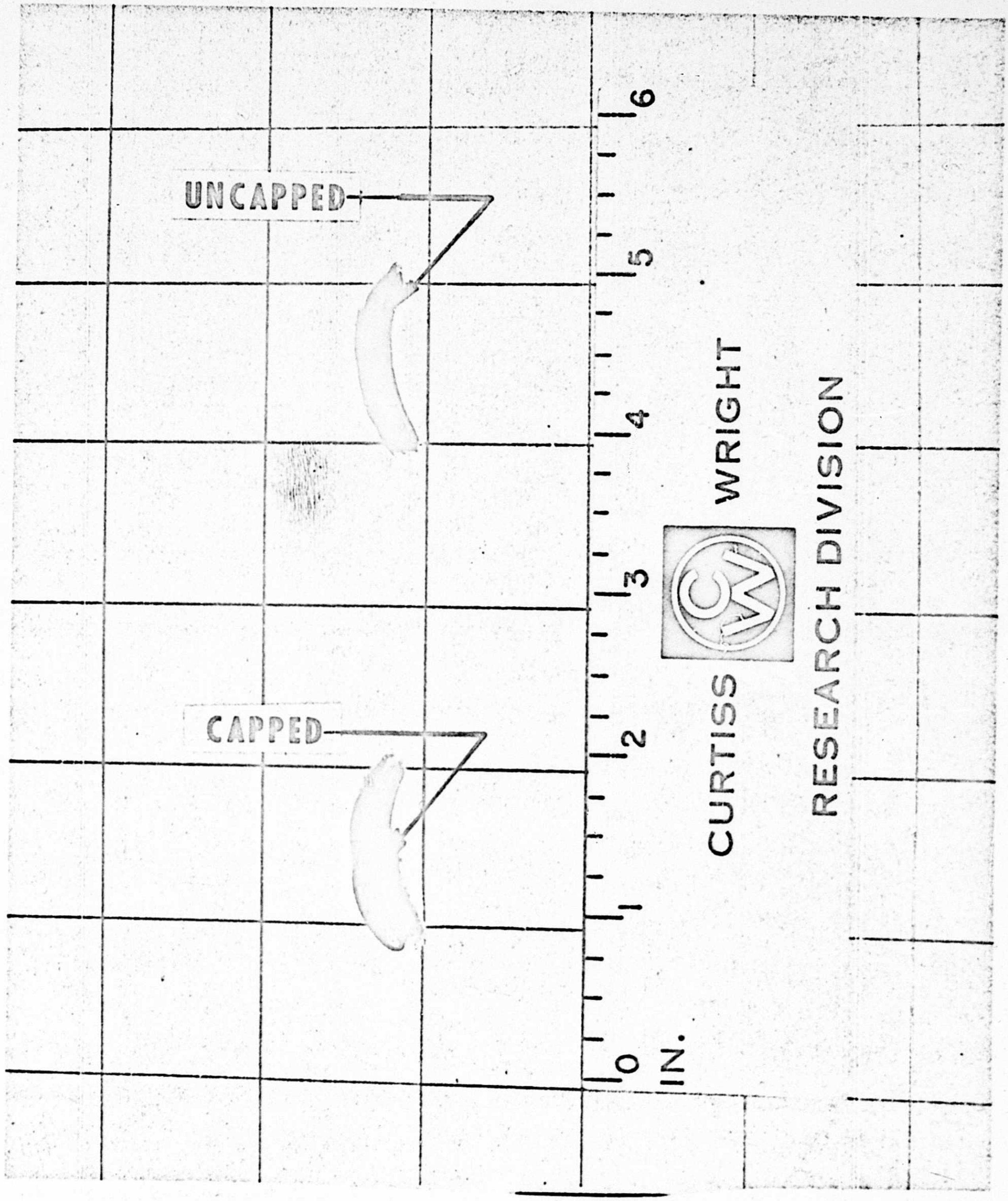
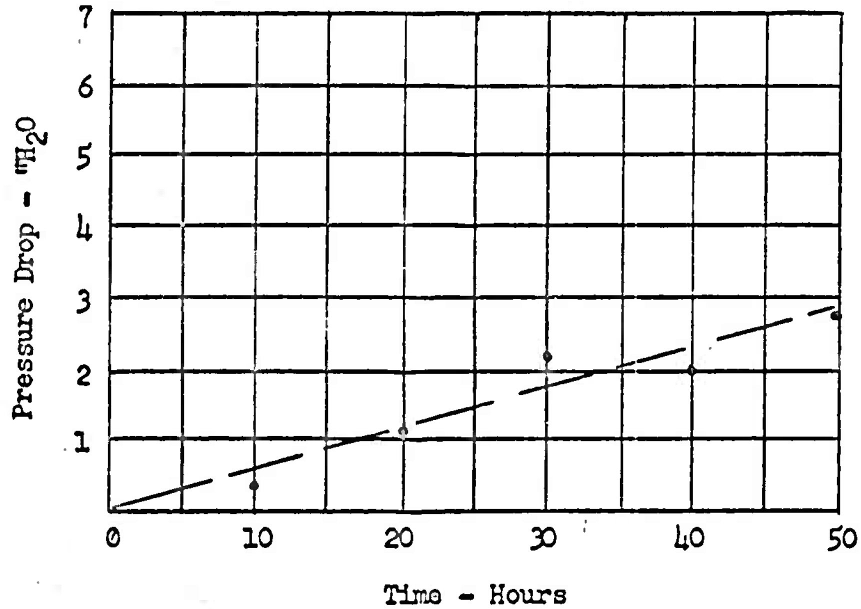


FIG.6—END VIEW - AFTER TEST

TRANSPIRATION COOLED TURBINE BLADE

CONTAMINATION TEST

(a) Pressure Drop vs Time



(b) Flow Loss vs Time

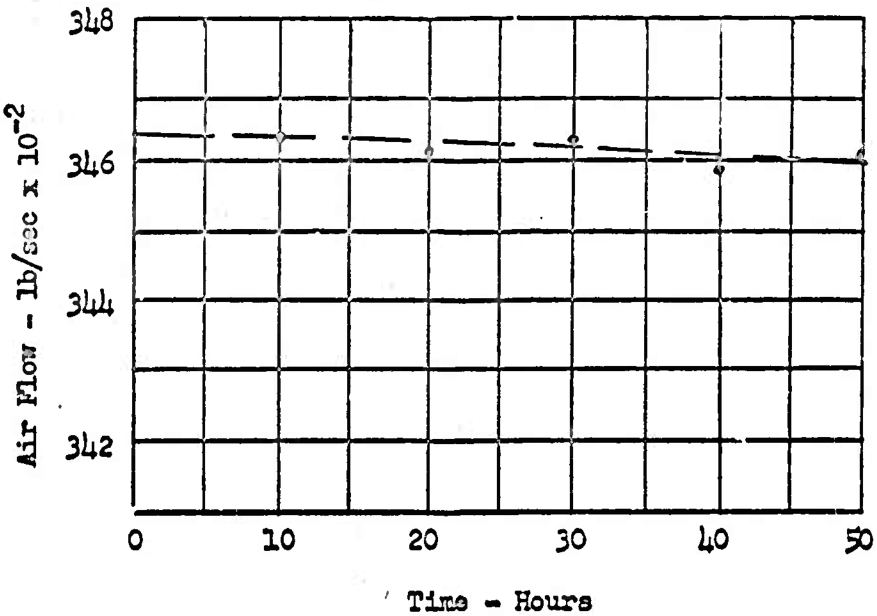
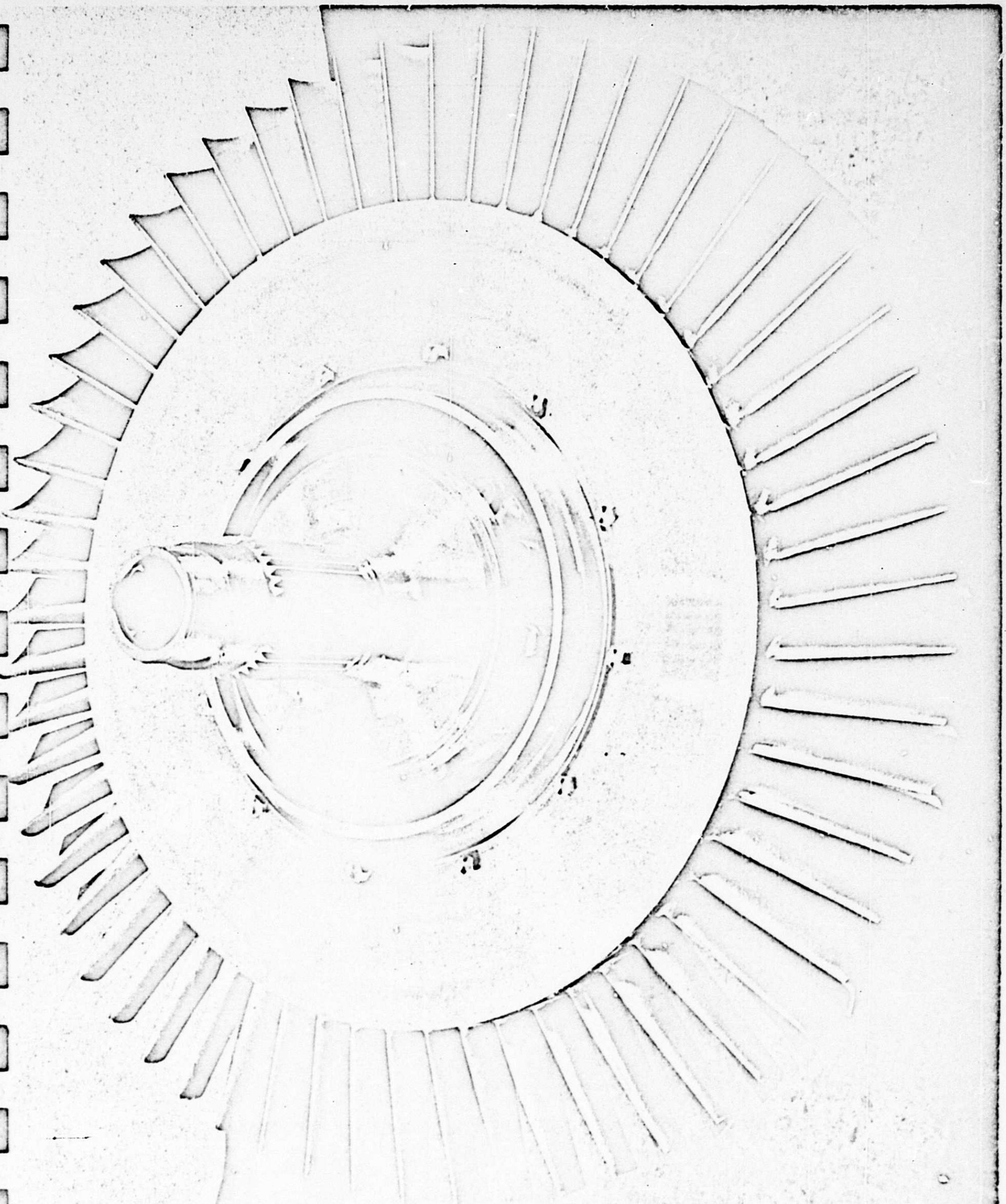
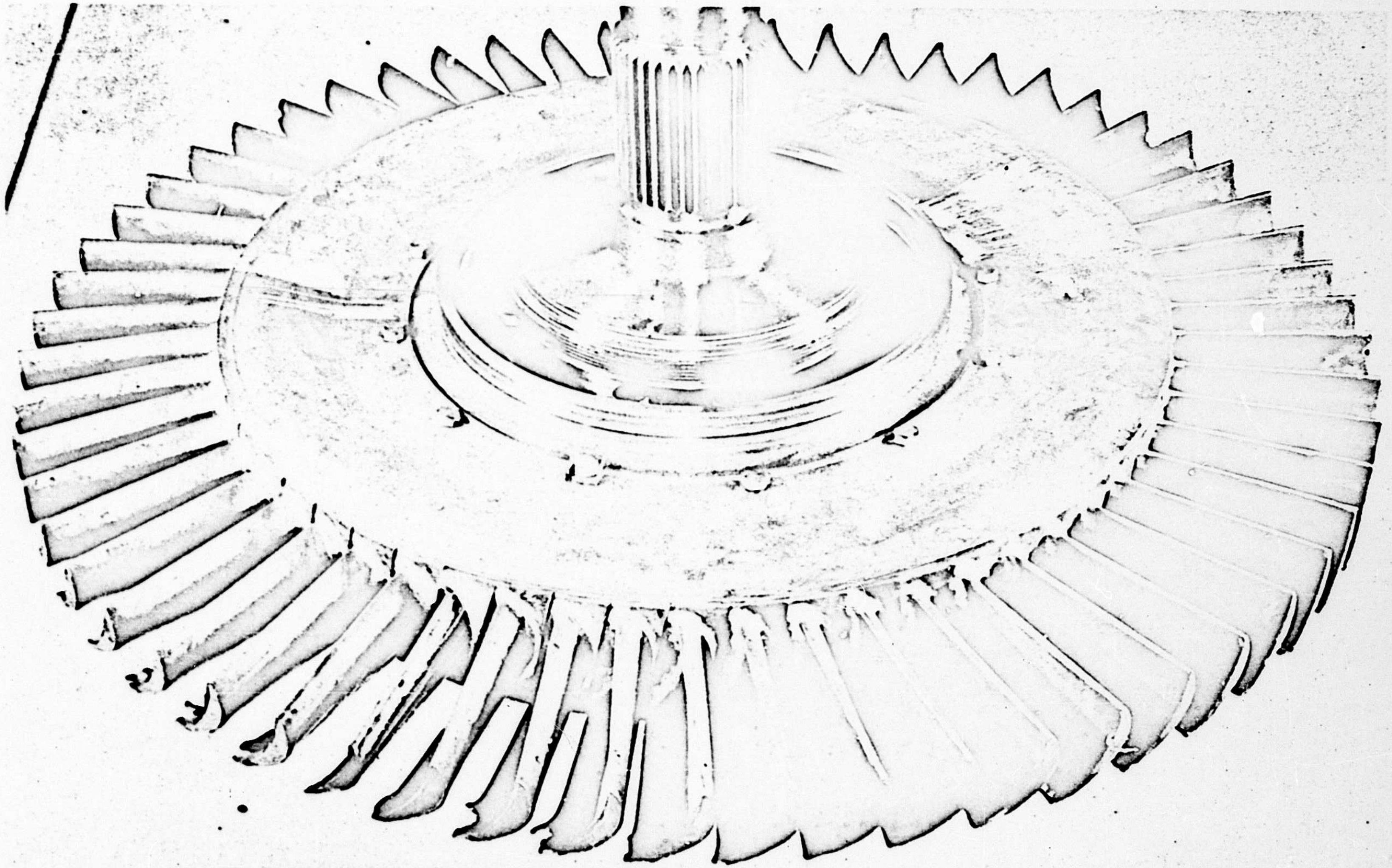


Figure VI-7



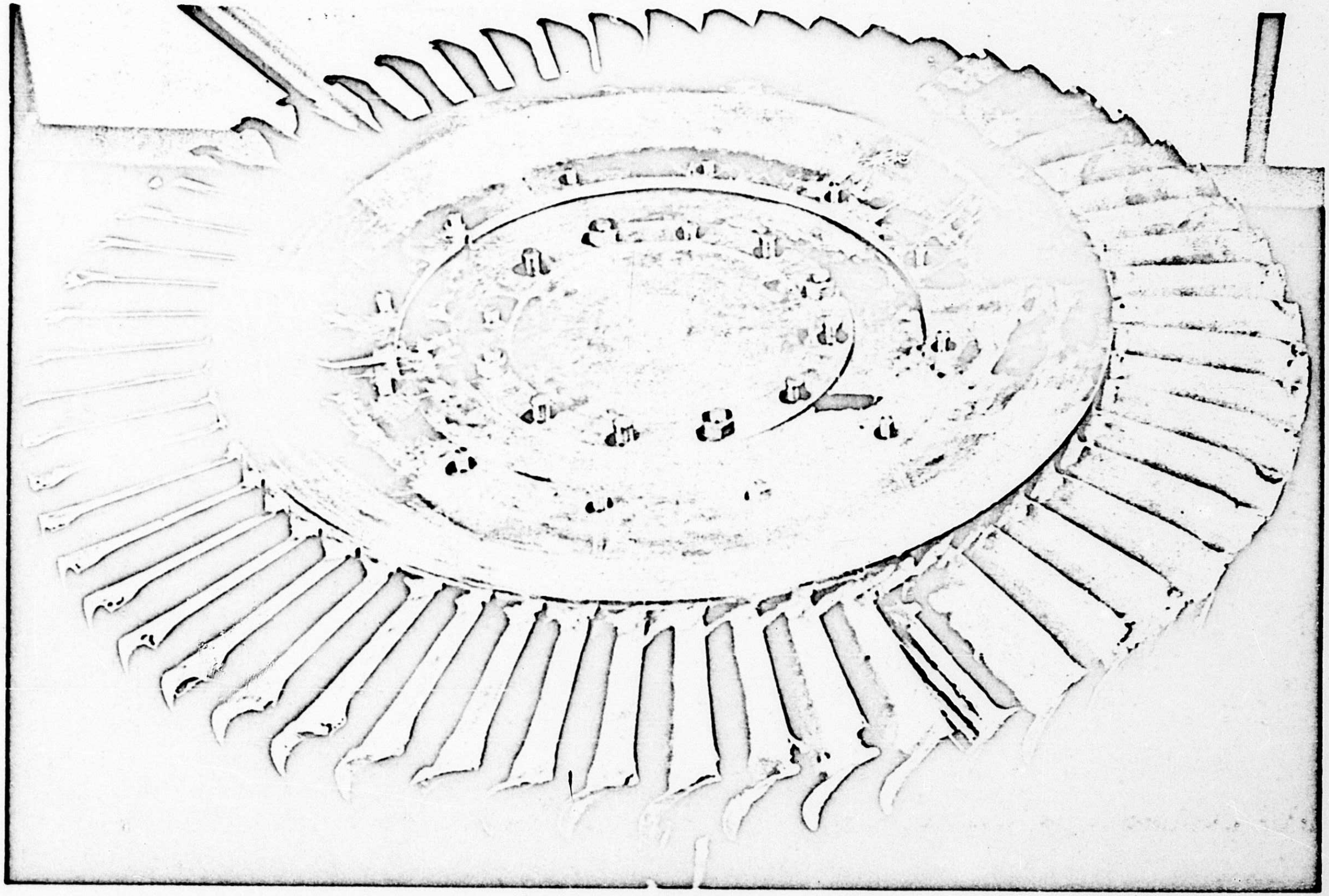
TRANSPIRATION COOLED TURBINE ROTOR ASSEMBLY

Figure VI-8



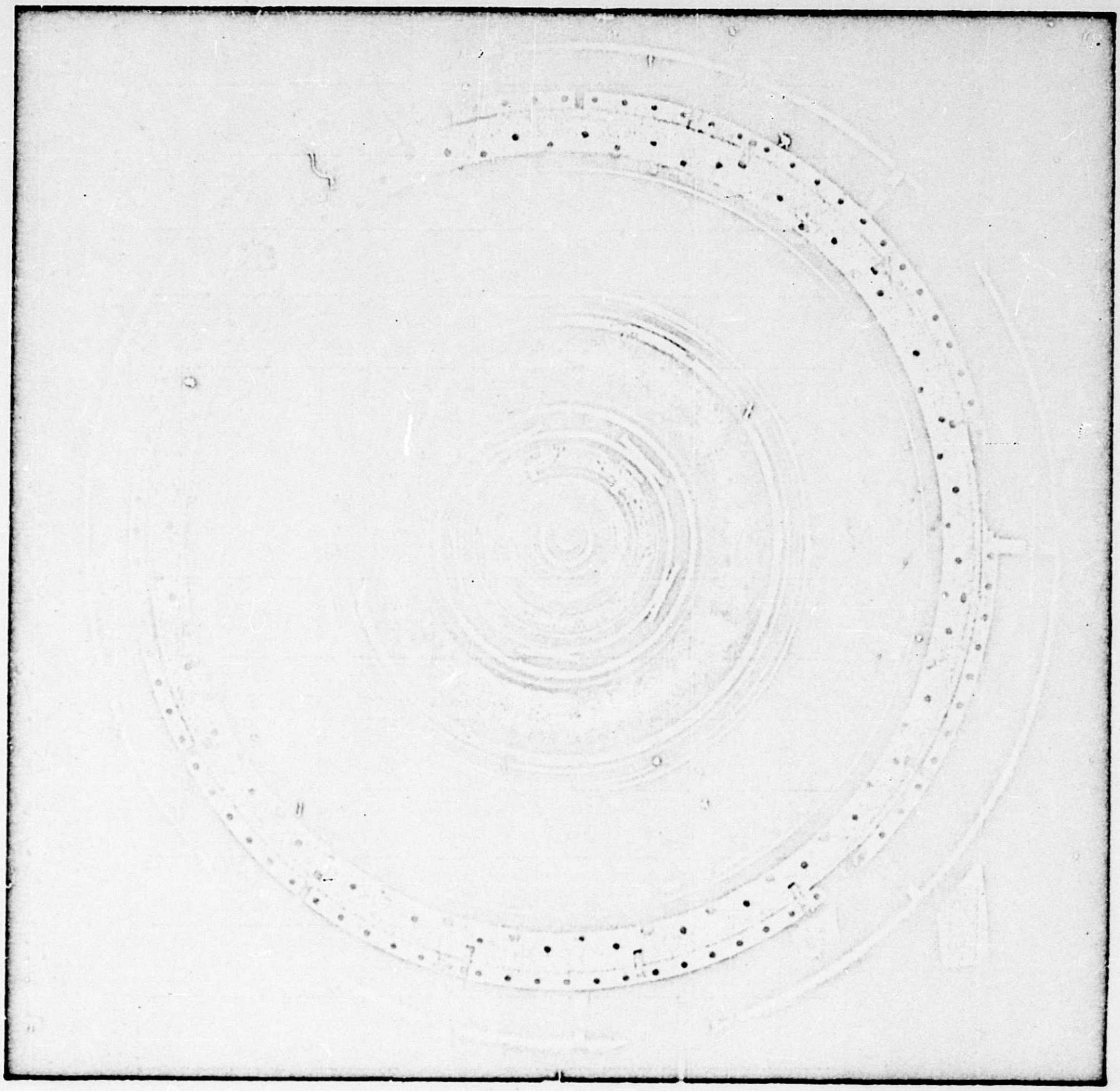
TRANSPIRATION COOLED TURBINE ROTOR ASSEMBLY
SHOWING BLADE WITH MISSING AIRFOIL AFTER TEST (FRONT)

Figure VI-9



TRANSPIRATION COOLED TURBINE ROTOR ASSEMBLY
SHOWING BLADE WITH MISSING AIRFOIL AFTER TEST (REAR)

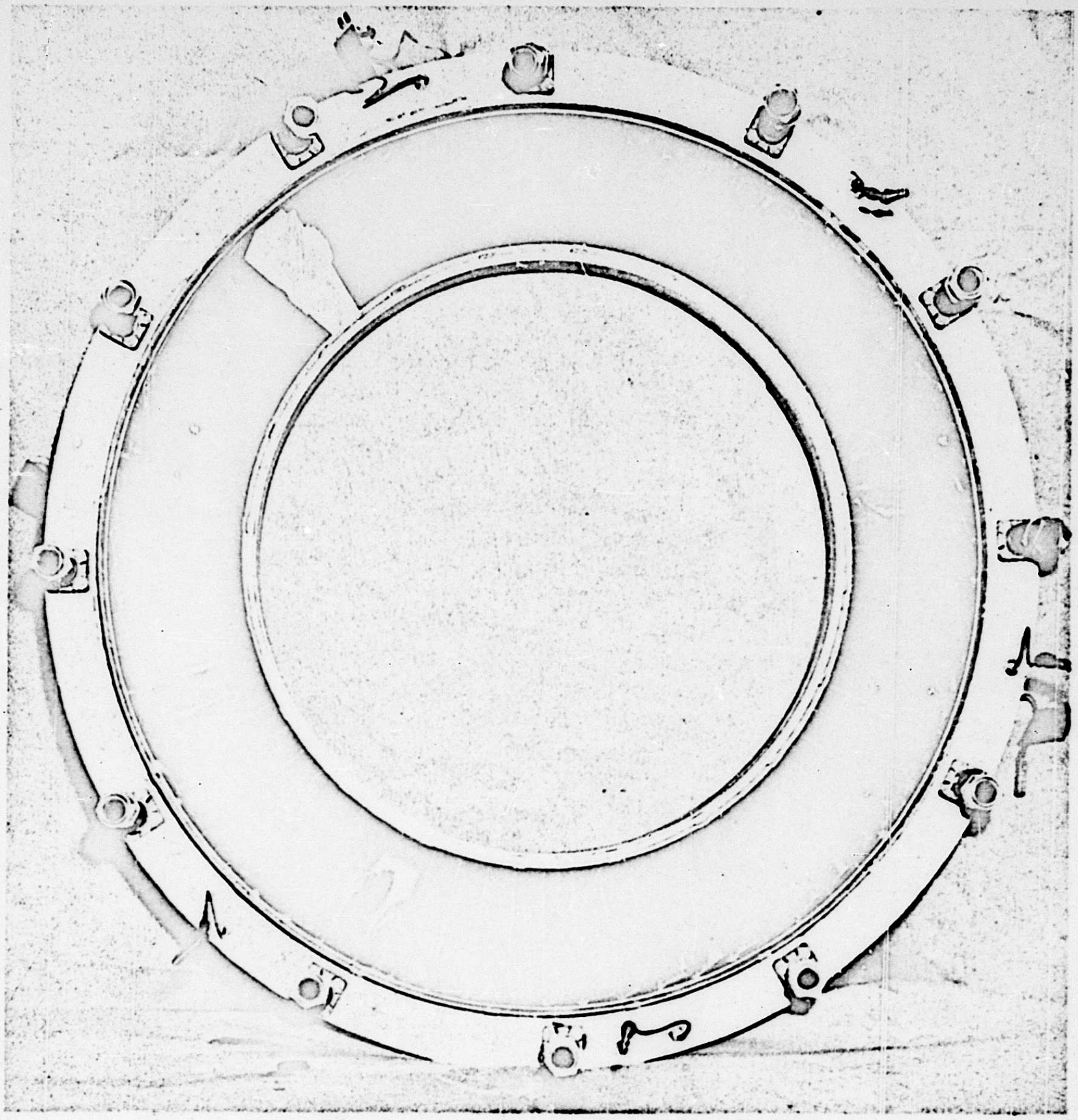
Figure VI-10



TRANSPIRATION COOLED STATOR BLADE ASSEMBLY
AFTER COMBUSTION CHAMBER TORCHING CONDITION (REAR)

Figure VI-11

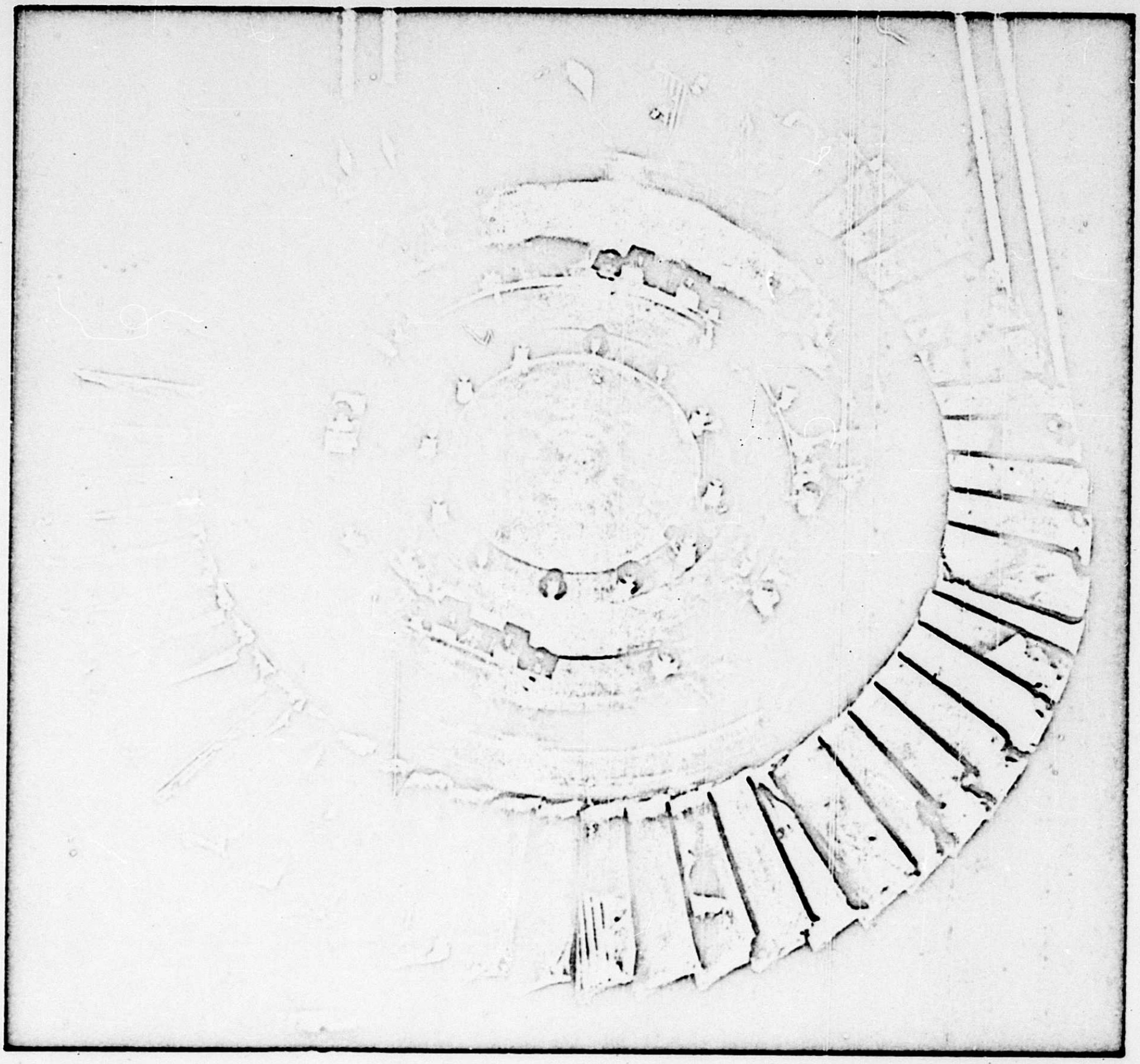
[redacted]



TRANSPIRATION COOLED STATOR BLADE ASSEMBLY
AFTER COMBUSTION CHAMBER TORCHING CONDITION (FRONT)

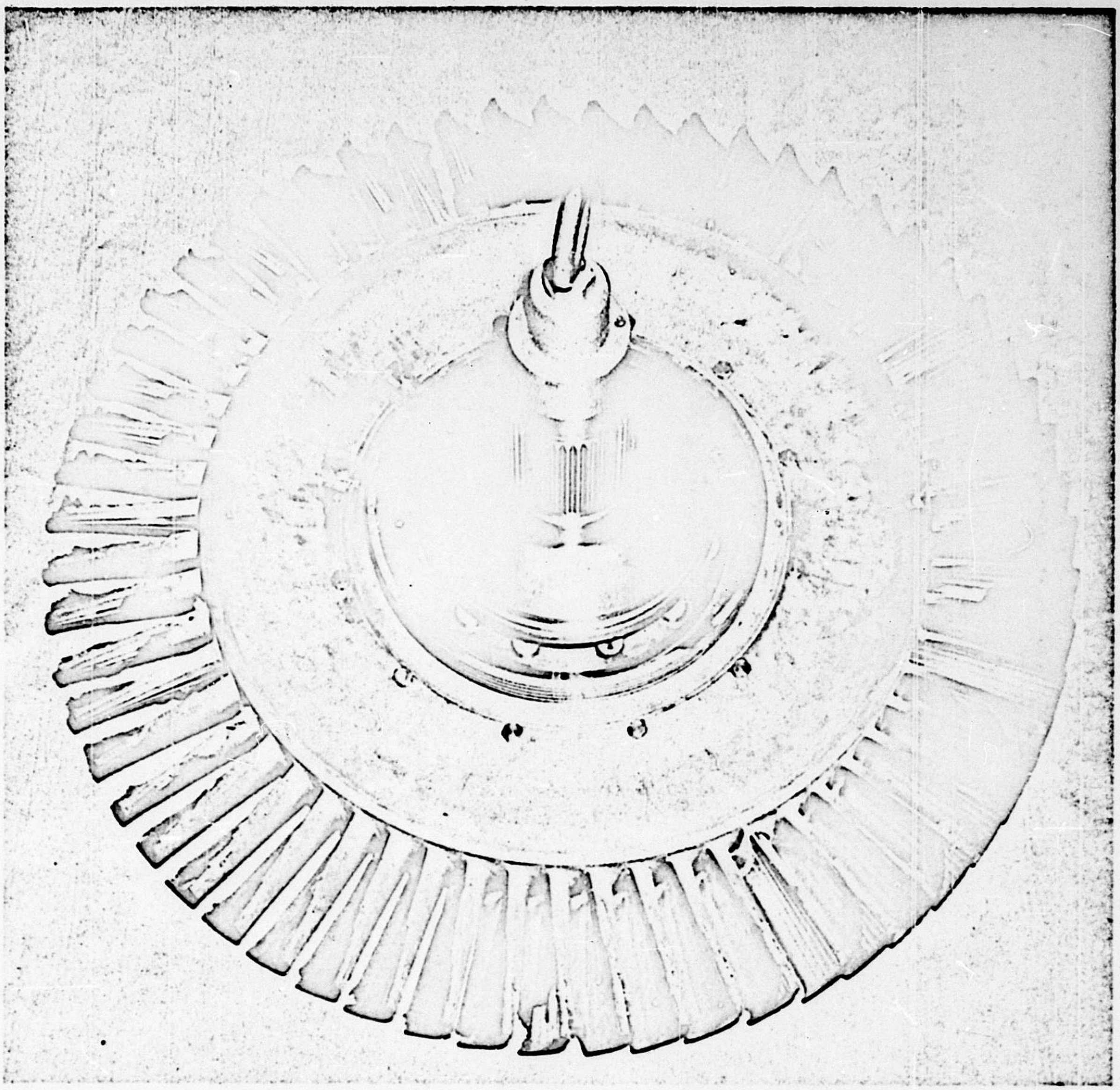
Figure VI-12

[redacted]



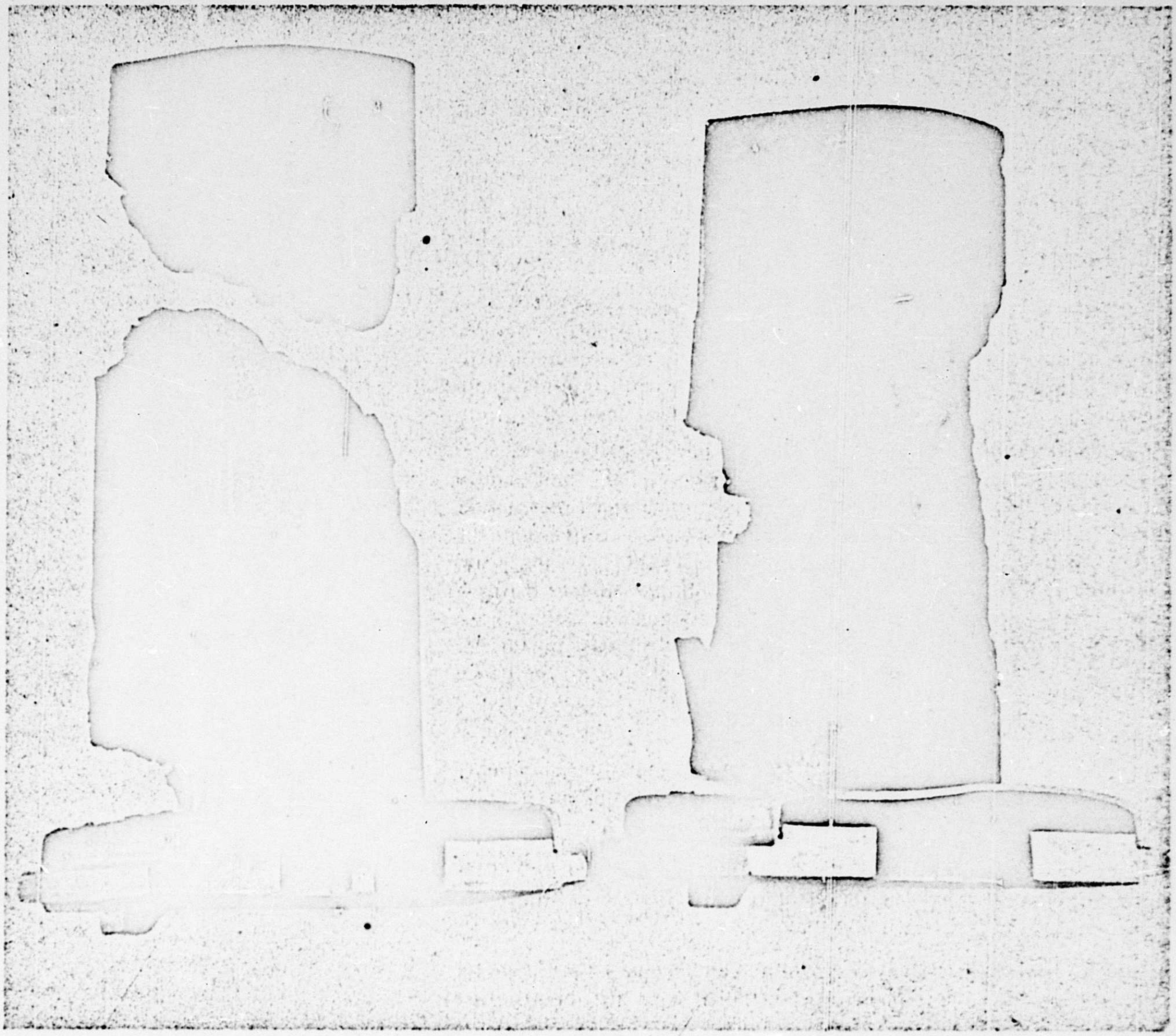
TRANSPIRATION COOLED TURBINE ROTOR ASSEMBLY
SHOWING DAMAGES RESULTING FROM BURNED STATOR (REAR)

Figure VI-13



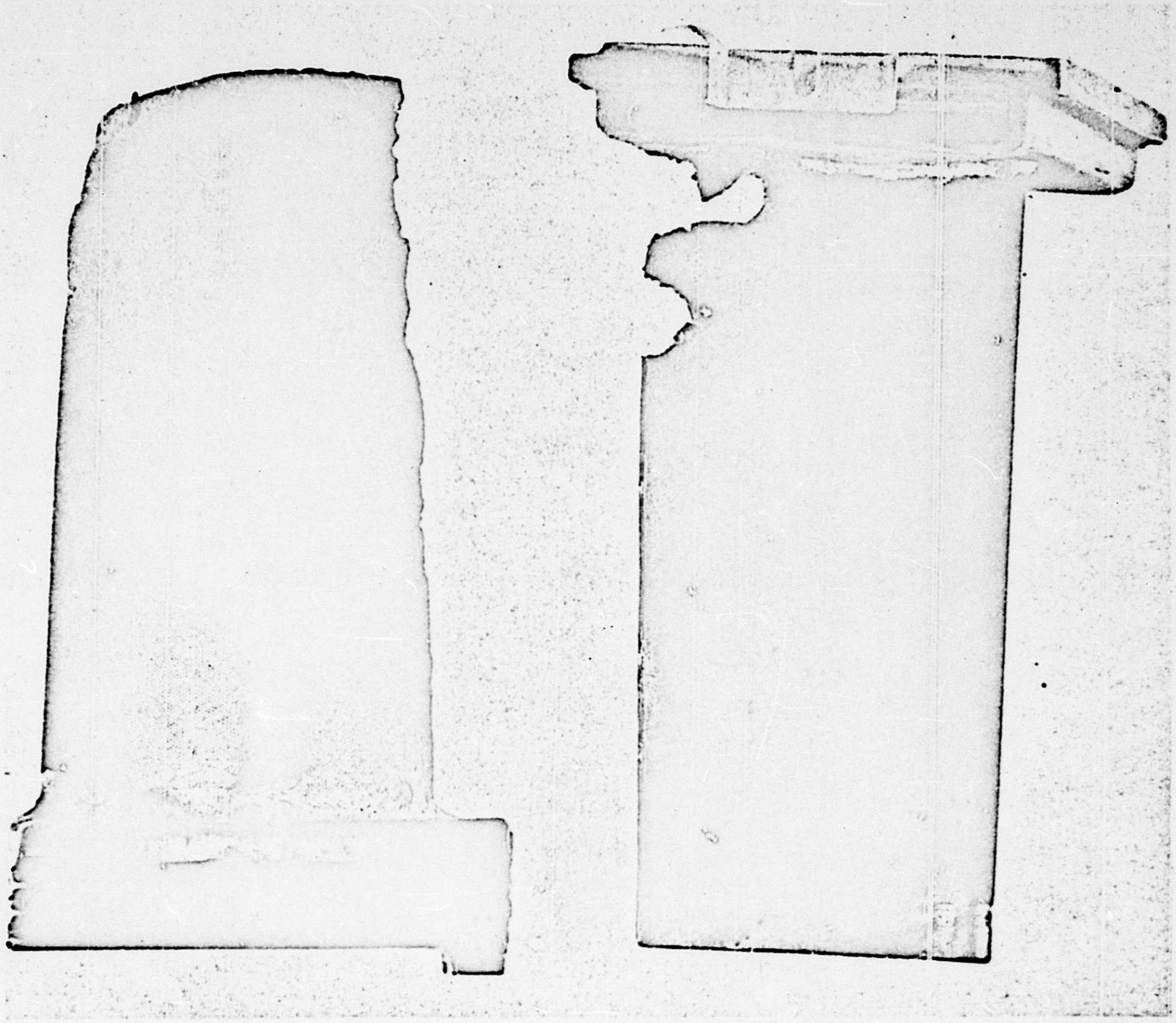
TRANSPIRATION COOLED TURBINE ROTOR ASSEMBLY
SHOWING DAMAGES RESULTING FROM BURNED STATOR (FRONT)

Figure VI-14



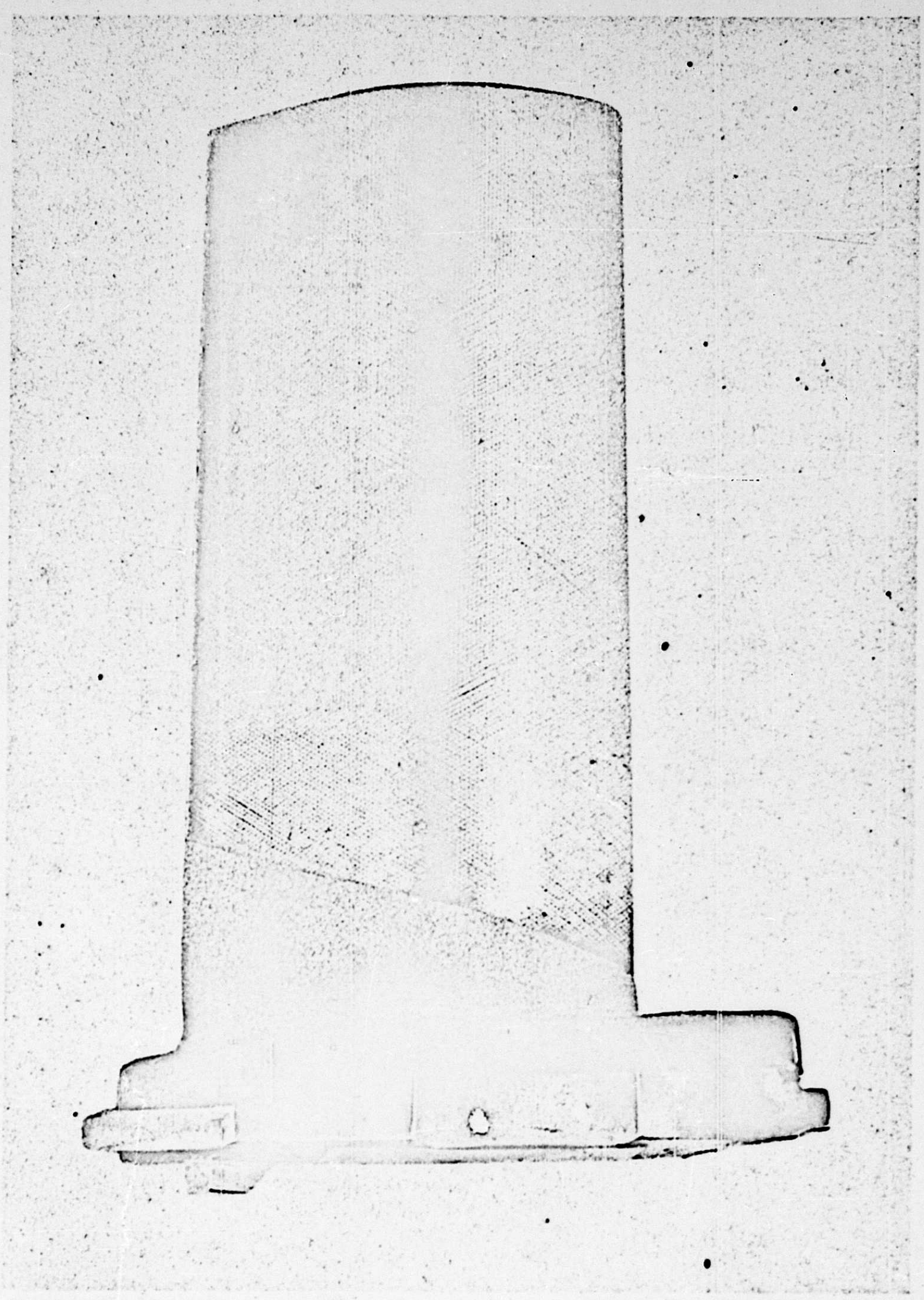
TRANSPIRATION COOLED STATOR BLADE
FAILED BY COMBUSTION CHAMBER TORCHING AND COOLING AIR STARVATION

Figure VI-15



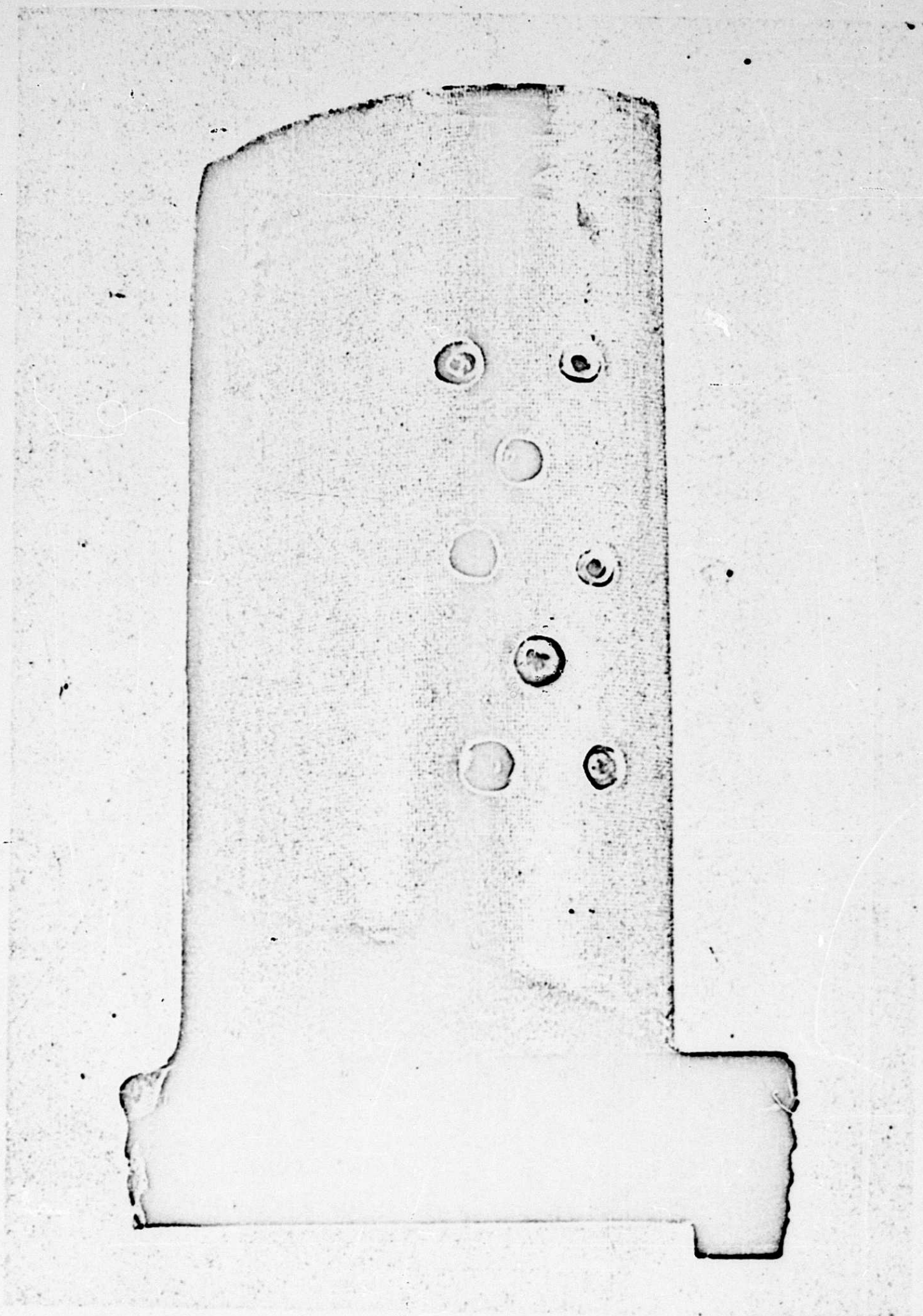
TRANSPIRATION COOLED TURBINE BLADES
SECONDARY DAMAGE FROM BURNED STATOR BLADE

Figure VI-16



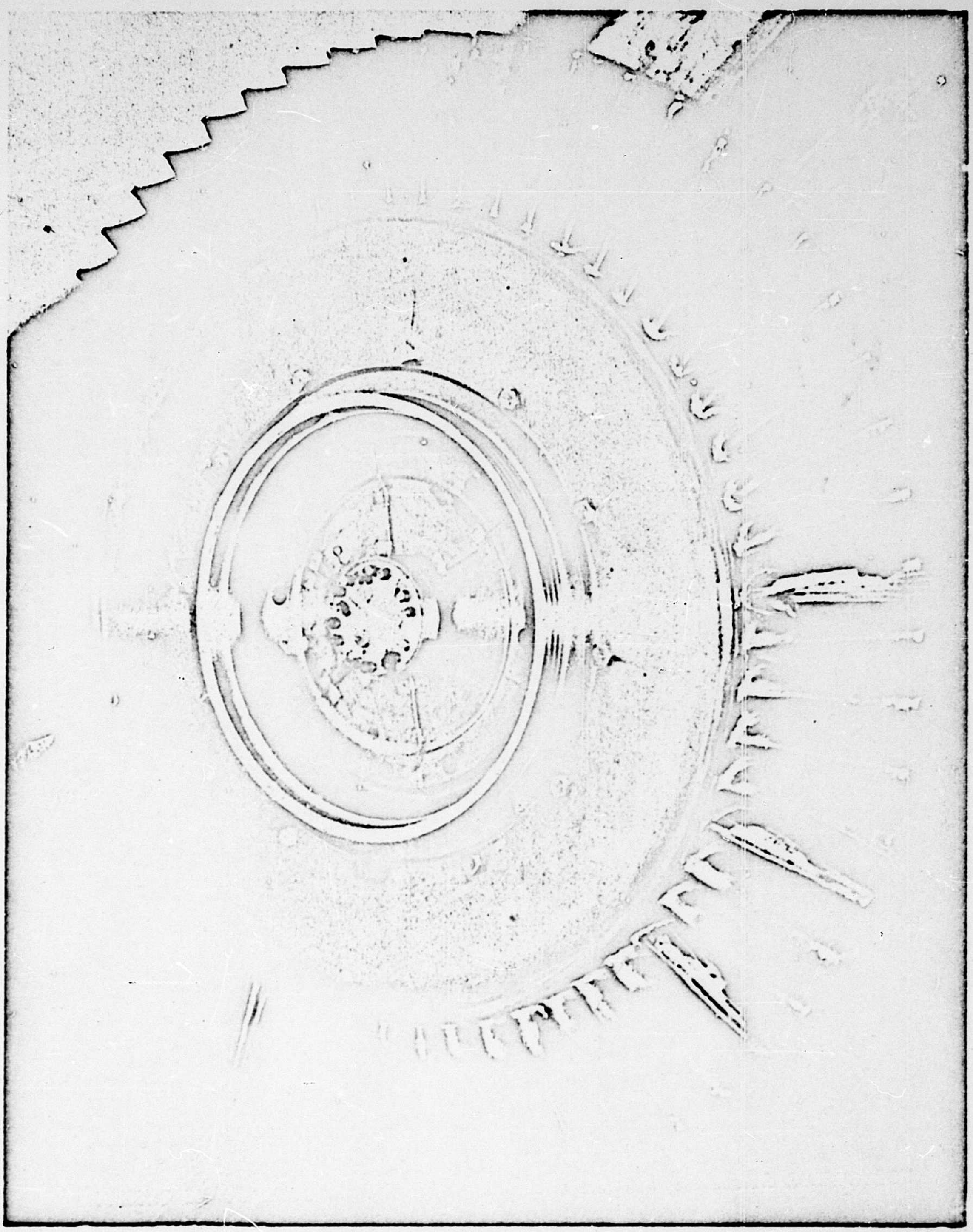
**TRANSPIRATION COOLED STATOR BLADE
METALIZED ON THE SUCTION SIDE AT THE ROOT**

Figure VI-17



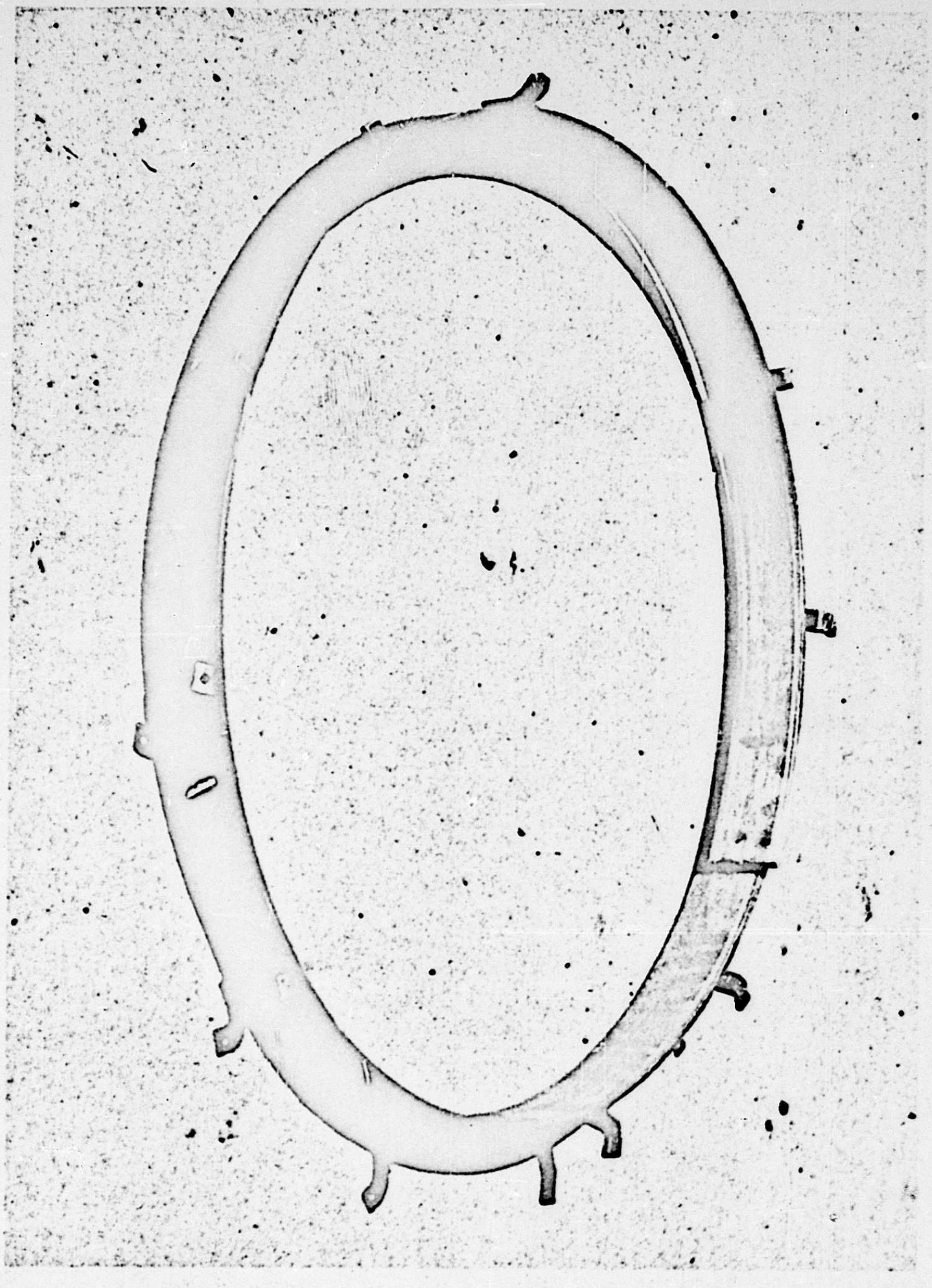
TRANSPIRATION COOLED ROTOR BLADE
PLUG WELDED AIRFOIL AND METALIZED ROOT

Figure VI-18



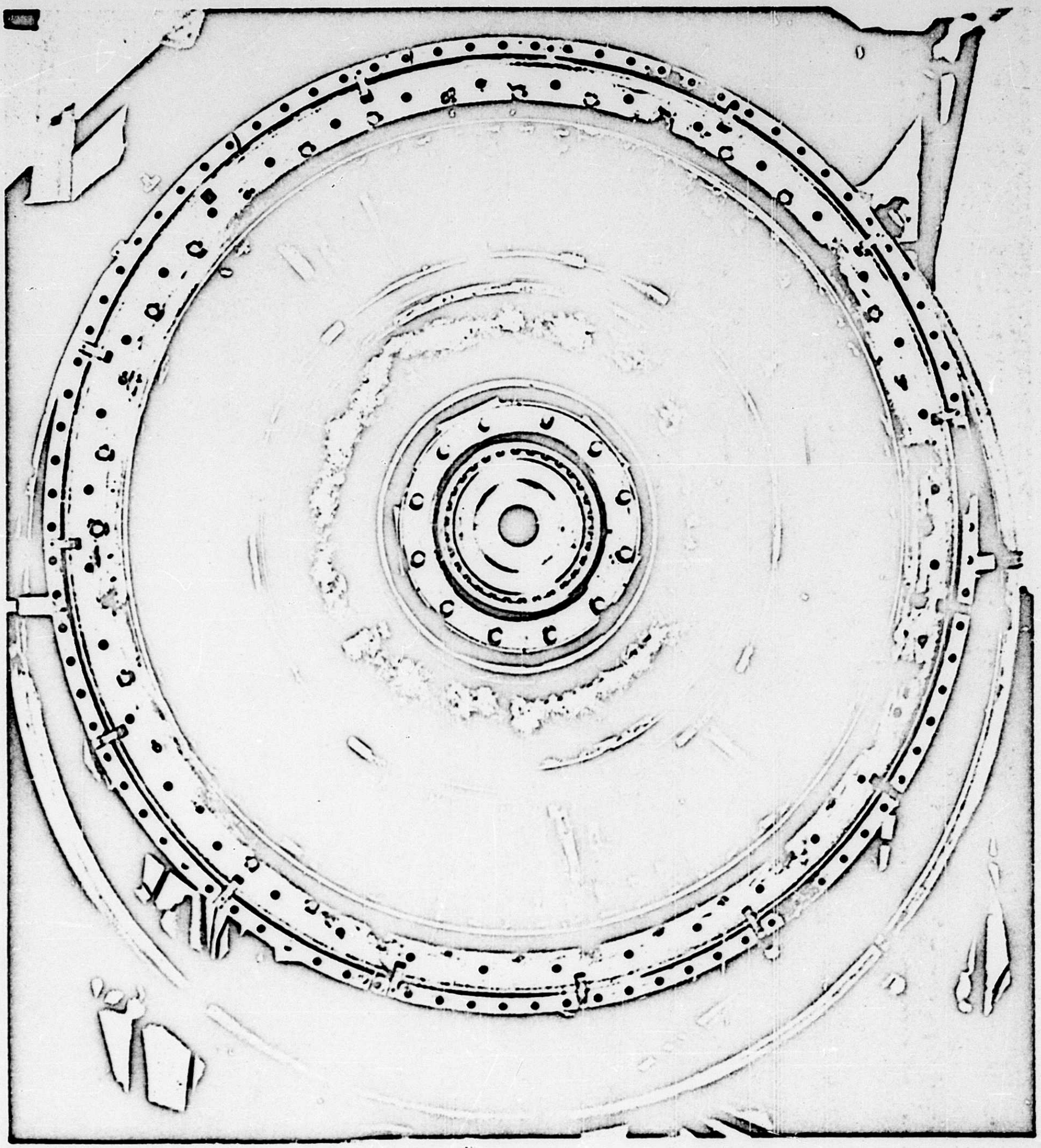
TRANSPIRATION COOLED ROTOR ASSEMBLY
SHOWING DAMAGE INCURRED IN LAST TEST

Figure VI-19



FAILED TURBINE ROTOR SHROUD

Figure VI-20



TRANSPIRATION COOLED STATOR BLADE ASSEMBLY
AFTER FAILURE OF TURBINE SHROUD RING

Figure VI-21

2500°F TRANSPIRATION COOLED TURBINE
TEST VEHICLE PERFORMANCE DATA

Gas Temperatures
With VAN

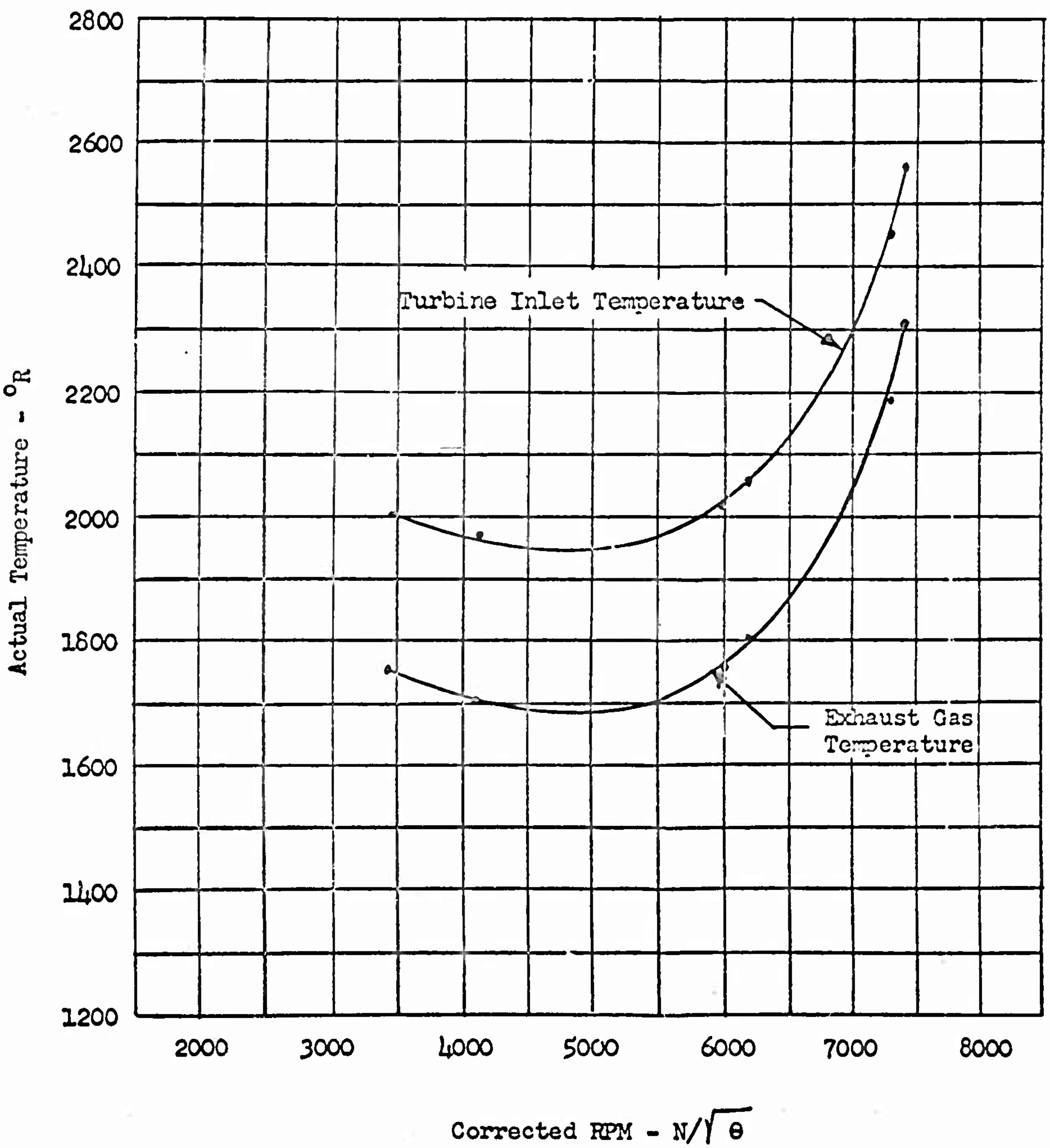


Figure VI-22

2500°F TRANSPIRATION COOLED TURBINE

TEST VEHICLE PERFORMANCE DATA

Gas Temperatures

Without VAN

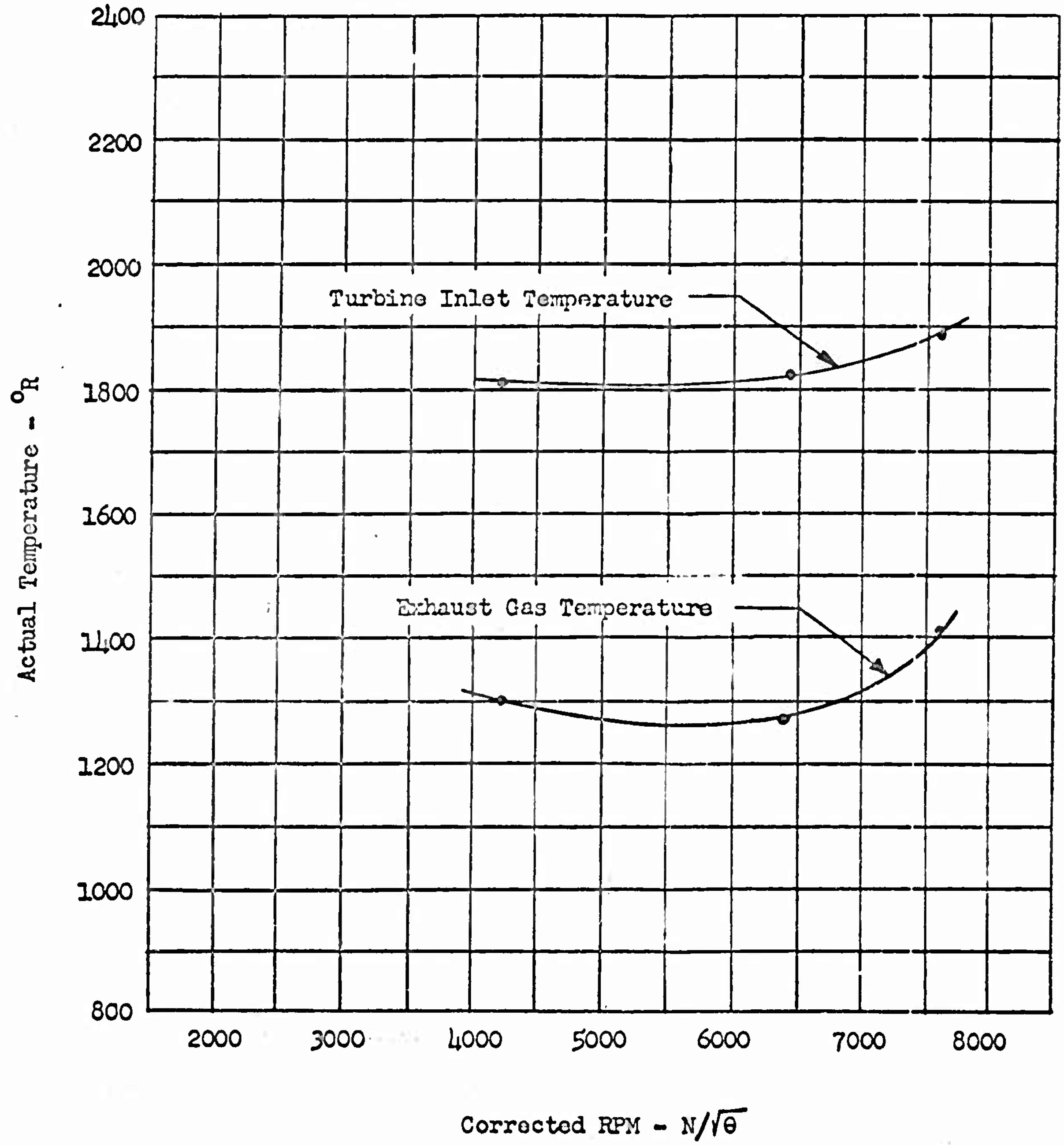
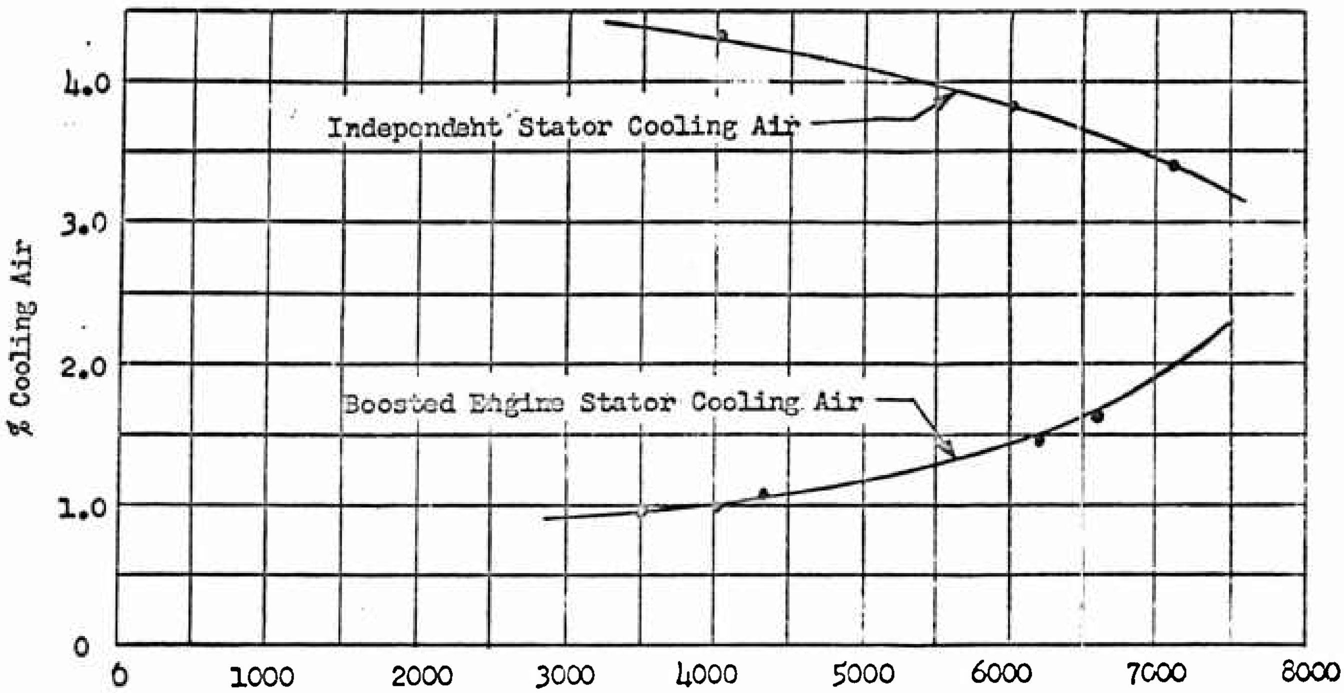


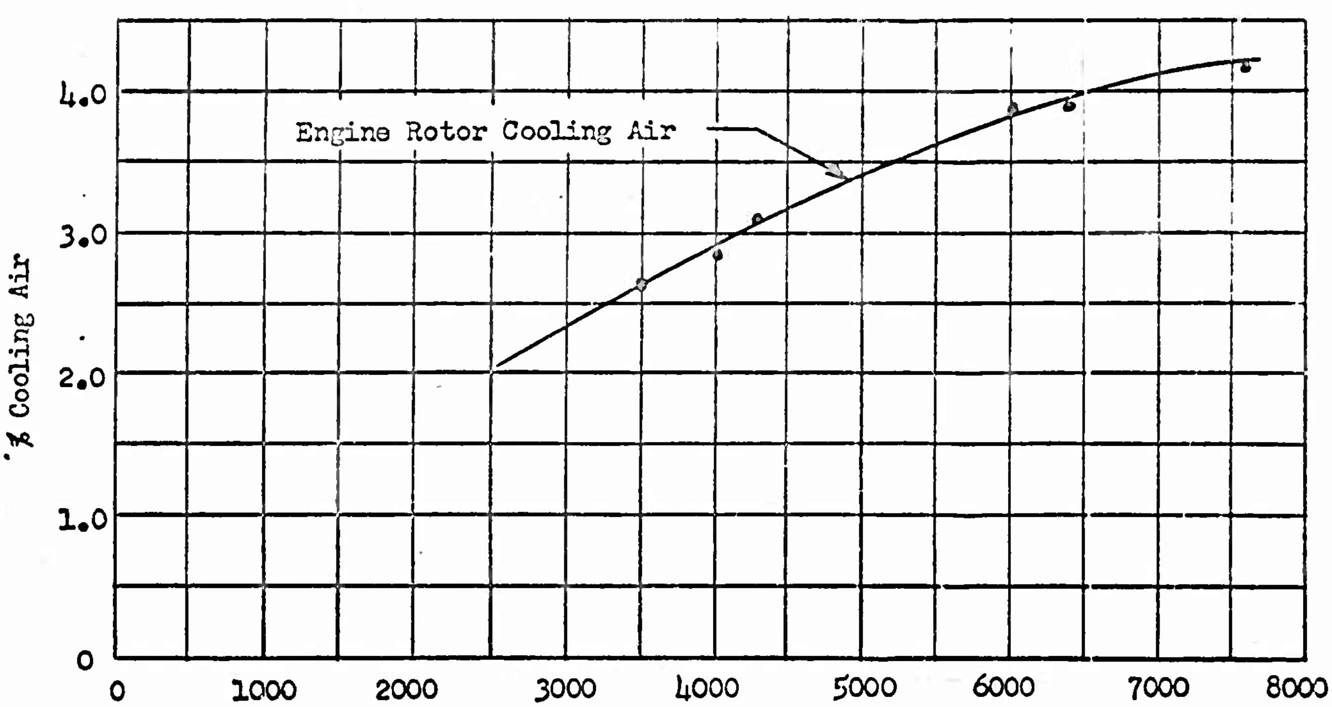
Figure VI-23

2500°F TRANSPIRATION COOLED TURBINE

BLADE COOLING AIR FLOWS



(a) Corrected RPM - $N/\sqrt{\theta}$



(b) Corrected RPM - $N/\sqrt{\theta}$

Figure VI-24

2500°F TRANSPIRATION COOLED TURBINE

AIR TEMPERATURES

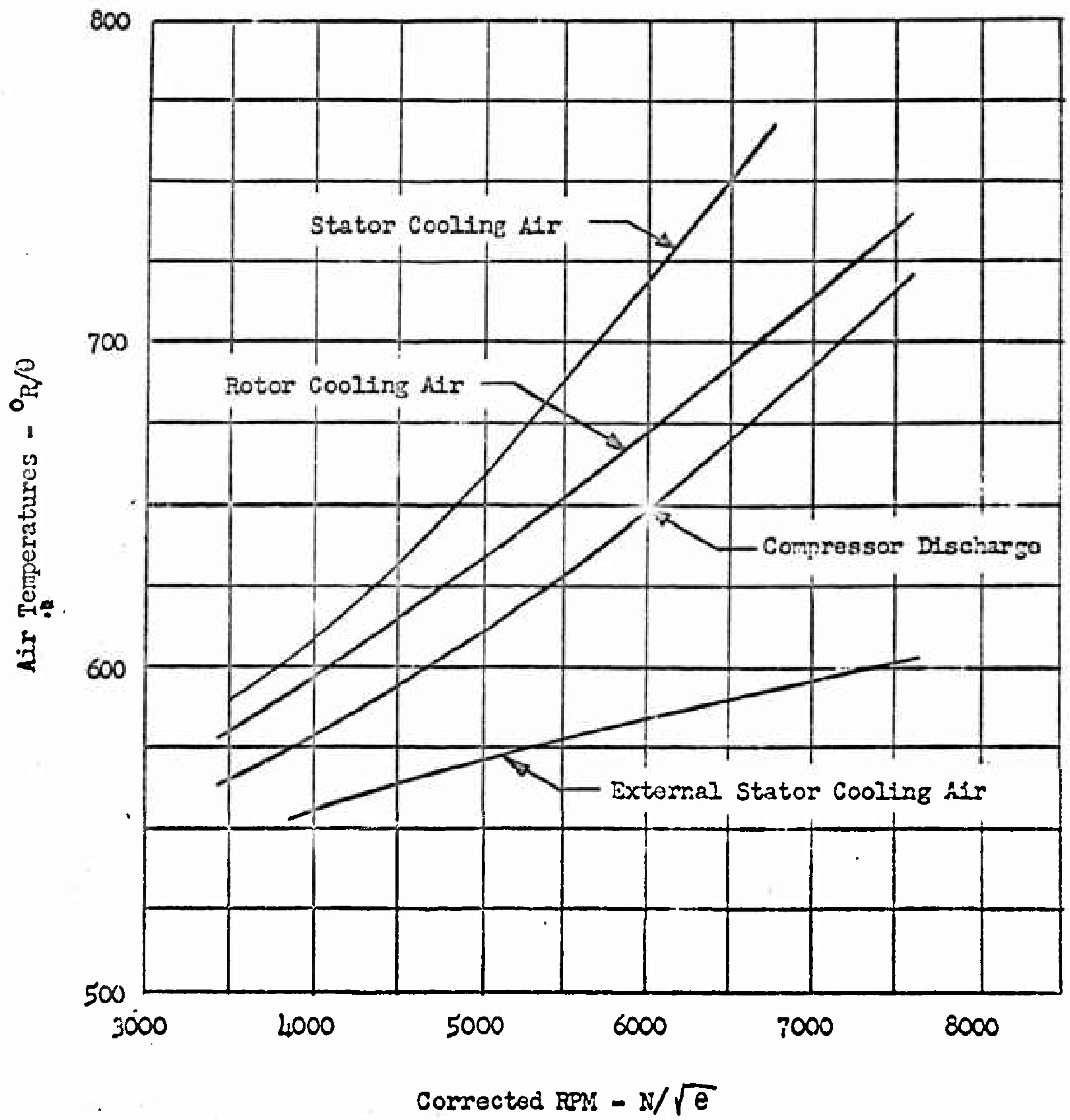


Figure VI-25

2500°F TRANSPIRATION COOLED TURBINE

COMPRESSOR AIR FLOW

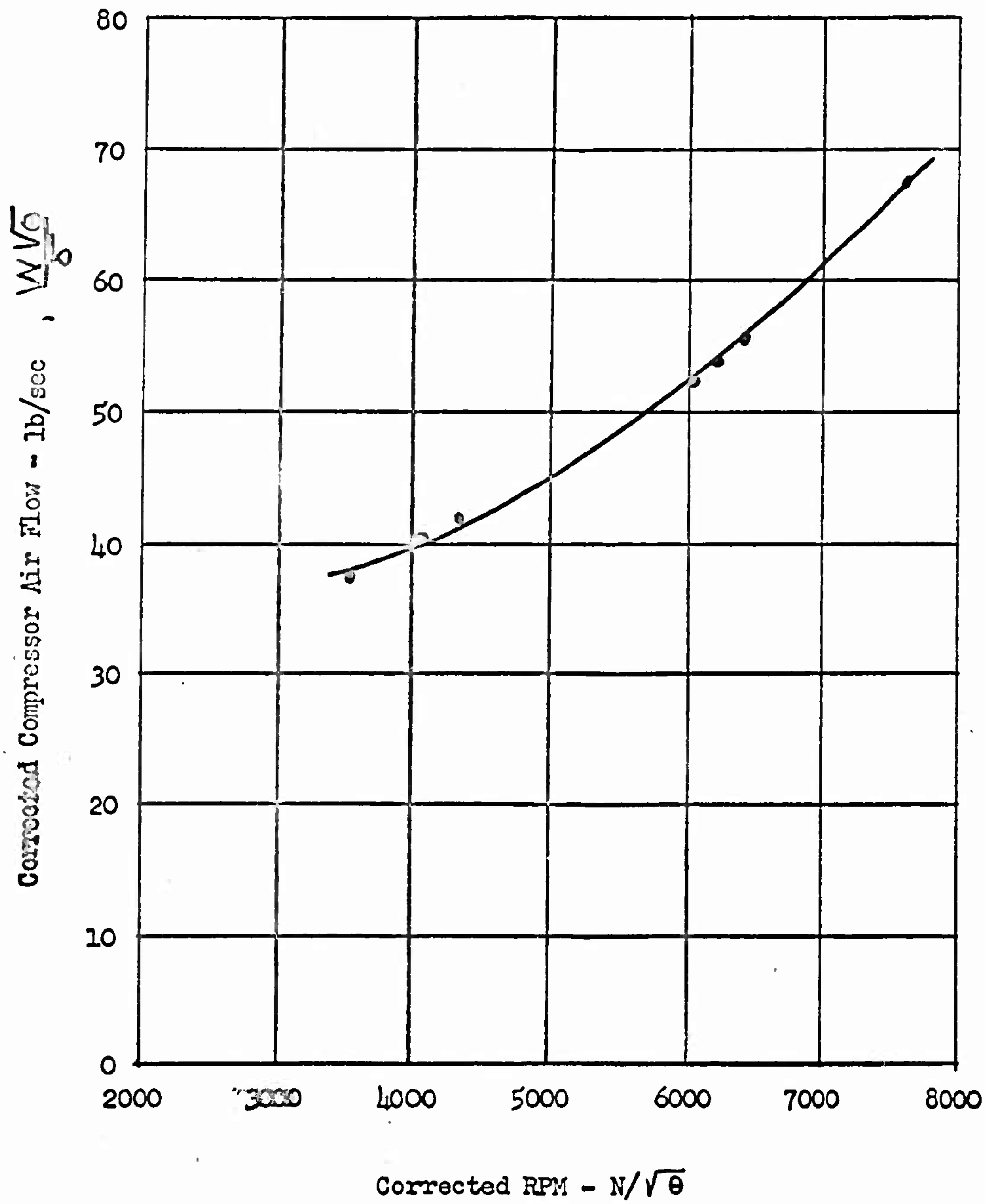


Figure VI-26

2500°F TRANSPIRATION COOLED TURBINE TEST VEHICLE PERFORMANCE DATA

PRESSURES

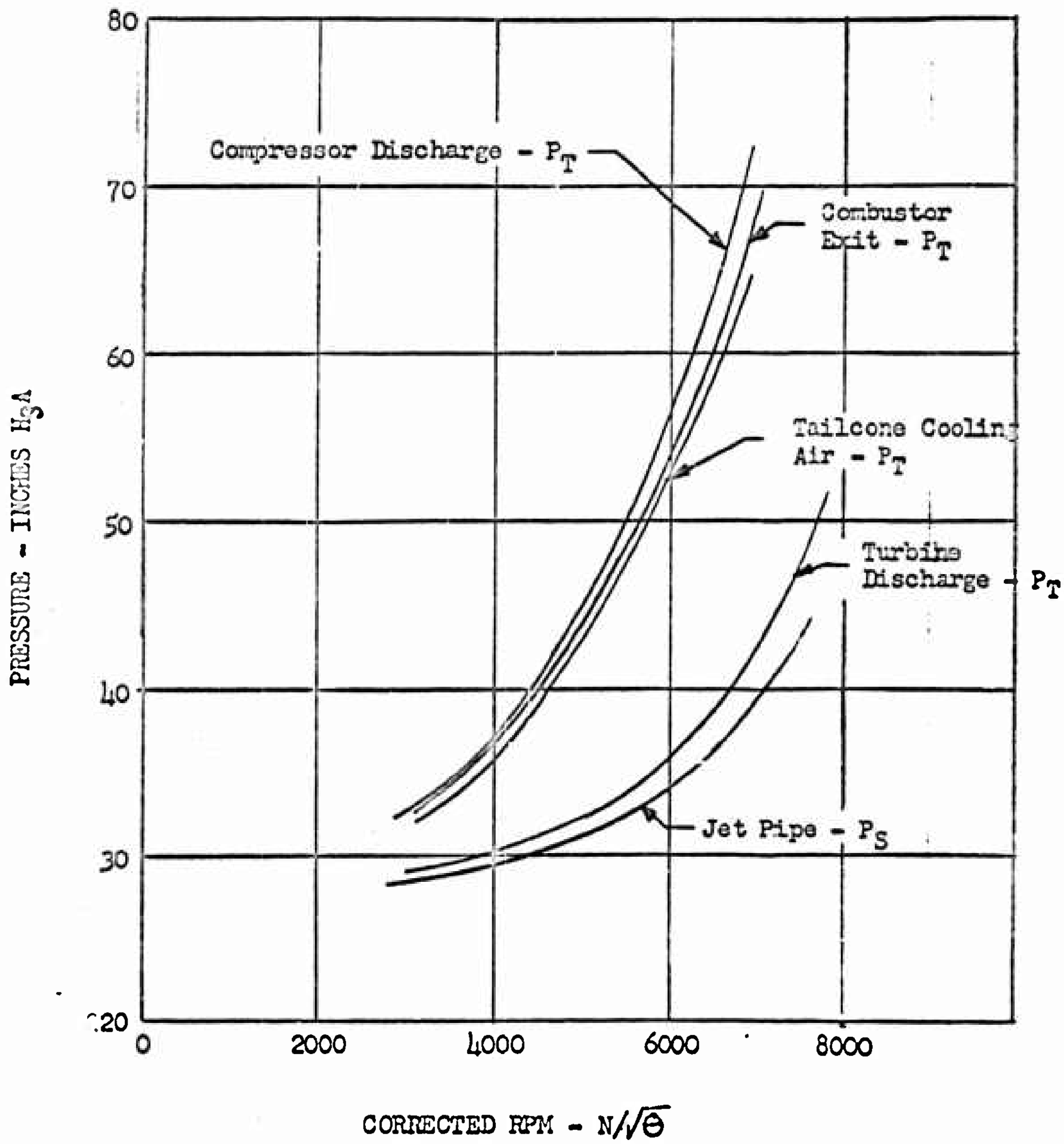


Figure VI-27

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